THE NATIONAL BOILER & GENERAL INSURANCE COMPANY LIMITED
Head Offices: 22 St. Ann's Sq.r Manchester.

Telegraphic Address
"NATIONAL," MANCHESTER
Telephone No. 1409

NOTES

ON

Material, Construction and Design

OF

LAND BOILERS

BY

EDWARD G. HILLER, B.Sc., M.I.C.E., M.I.Mech.E.

Chief Engineer,

THE NATIONAL BOILER AND GENERAL INSURANCE COMPANY LIMITED,

22, St. Ann's Square, Manchester.

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1907.

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IMPROVED VALVE SEATED PLUG.

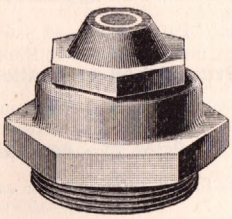

FIG. 6.

Low-pressure Type.

Screwed 1 in., 1¼ in., 1½ in., 2 in.

Gas Thread on the Seating.

IMPROVED VALVE SEATED PLUG.

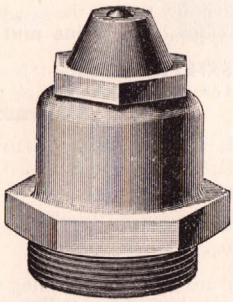

FIG. 8.

High-pressure Type.

Screwed 1 in., 1¼ in., 1½ in., 2 in.

Gas Thread on the Seating.

IMPROVED SCREW CONE LOCO. PLUG.

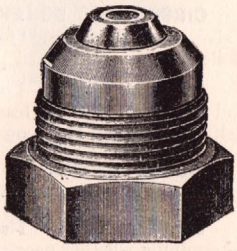

FIG. 12.

For Boilers of the Locomotive Type.

Screwed 1 in., 1¼ in., 1½ in.

Gas Thread on the Seating.

For further particulars see detailed Price List.

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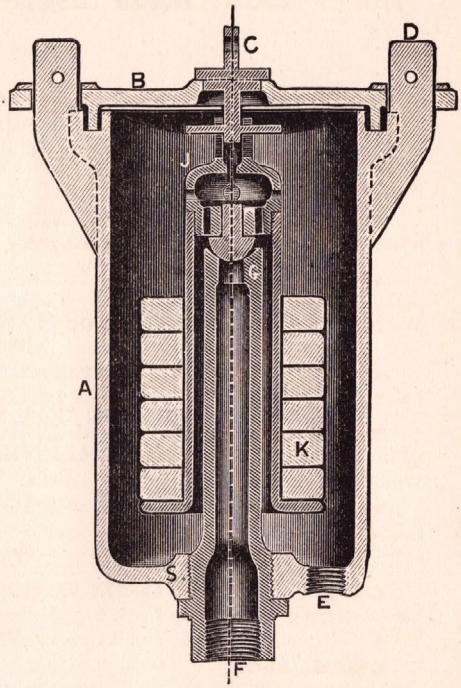

RECOMMENDED by the London County Council and other public bodies.

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NOTES ON MATERIAL, CONSTRUCTION, AND DESIGN OF LAND BOILERS.

SECOND EDITION.

The following notes have been issued in response to numerous enquiries from Boiler Makers and others, and they embody the principal features of the practice of the "National" Company in the design of Land Boilers.

The Rules given are based on the experience of the "National" Company, and on experiments by various authorities.

They are given here as recommendations only, and are not a complete set of Rules for Boiler design, and while they will often be of assistance they do not dispense with the necessity of submitting the design to us where it is specified that the Boiler is to be to our satisfaction.

Illustrations are given of the general arrangement of various types of Boilers in common use. (*See* list on preceding page.)

In this Second Edition some modifications have been made (*see* page 3) and the whole of the notes have been revised.

EDWARD G. HILLER,
Chief Engineer.

January 1907.

These notes are subject to revision from time to time as required.

I.

GENERAL NOTES AND TESTS OF MATERIAL.

THE following notes apply to Steam Boilers of ordinary types and sizes such as are generally used for manufacturing purposes ashore. They are not intended to apply to the design of Marine Boilers for seagoing purposes. RANGE OF APPLICABILITY OF RULES.

It is throughout assumed that Mild Steel, made by the acid open-hearth process, will be used, except when otherwise stated. MILD STEEL.

The rules refer to conditions such as exist in ordinary Steam Boilers where the plates and other parts may be exposed to the deteriorating influence of corrosion, and are subjected to heat. CONDITIONS THOSE AS IN STEAM BOILERS.

In the case of Air Receivers and similar vessels, if there is no risk of corrosion the parts may generally be stressed 10 to 15 per cent higher than in a Steam Boiler under otherwise similar conditions. AIR RECEIVERS.

In order to give a good margin for wear and tear, it is advised that for fixed Boilers of ordinary sizes plates less than $\frac{3}{8}$ inch in thickness should not be used. MINIMUM THICKNESS OF PLATES.

It is advised that the measurements be taken in the case of Horizontal Boilers, such as Lancashire and Cornish Boilers, as follows:— MEASUREMENTS OF SIZE OF BOILERS.

Length—inside the end plates.

Diameter of shell—inside outer ring of plate.

Diameter of furnace and flue tubes—outside the ring of plate.

MATERIAL. MATERIAL.

As a rule, for the external shells of Internally-fired Boilers, such as Lancashire Boilers, and also for external shells of Boilers, such as some types of Water-tube Boilers, in which the shell is not subject to the first heat from the furnace, steel having an EXTERNAL SHELLS OF INTERNALLY-FIRED BOILERS.

ultimate tensile strength of 26 to 30 tons per square inch, and an elongation of not less than 20 per cent on 10 inches is advised.

Material of Flue Tubes and Ends of Externally-fired Shells.

For external shells of Externally-fired Boilers, Water-tube Boilers, plates into which tubes are to be expanded, all plates which are to be welded or flanged, and also for the furnace, flue tubes, and end plates of Lancashire and similar Boilers, steel having an ultimate tensile strength of 24 to 28 tons per square inch and an elongation of not less than 20 per cent on 10 inches is advised.

The tensile strength of the bars from which the rivets are made should be from 26 to 30 tons per square inch, with an elongation of not less than 25 per cent on a length of 8 inches.

TEST OF PLATES AND QUALITY OF MATERIAL.

TEST OF PLATES AND QUALITY OF MATERIAL.

The following specification and tests are recommended for the plates:—

All the plates, angles, and rivets to be of ductile steel of British manufacture, made by acid open-hearth process, free from laminations, pittings, or any other defects. The tensile strength of shell plates to be not less than 26 and not more than 30 tons per square inch. The furnace, flue tube, and end plates, and other plates as before mentioned, to have 24 to 28 tons' tensile strength, the elongation of all on lengths of 10 inches to be not less than 20 per cent, and to be, together with the tests thereof, entirely to the satisfaction of the Chief Engineer of The National Boiler and General Insurance Company Limited.

The brand of steel to be used to be submitted for approval.

Tests as follows must be made by the plate maker:—

SHELL PLATES.

Four cross test strips to be cut from each shell plate as follows:—

One from one end of the plate for tensile test.

One from same end of the plate for normal cold bending test.

One from opposite end of the plate for tensile test.

One from same end of the plate for heating and quenching bending test.

REMAINING PLATES.

Three test strips to be cut from each end plate, each flue and furnace plate, and each plate for gussets and butt straps in the part convenient to the plate maker, and to be tested as follows:—

One for tensile test.

One for normal cold bending test.

One for heating and quenching bending test.

The tensile and elongation tests to be in accordance with and to comply with the limits previously specified.

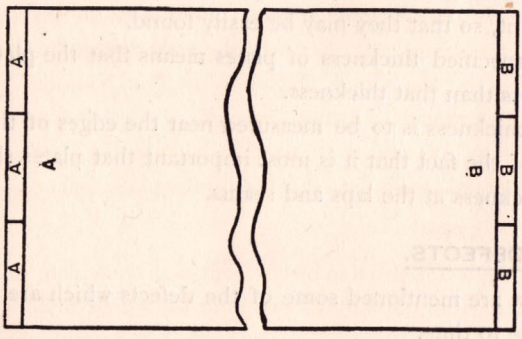
The normal cold bending test to be made by bending the strip when in its normal condition, and the heating and quenching bending test to be made after heating the strip to redness and slaking in water.

All strips for the bending tests shall stand bending without cracking until the sides are parallel, the inner radius being not greater than one and a half times the thickness of the plates.

POSITION OF TEST STRIPS.

On the annexed sketch is indicated, by the letters **A** and **B**, the best positions for taking the test strips of the shell plates, namely, across each end. The reason for this is that the ends of

Position of Test Strips.



the shell plates are more frequently defective than other parts, making it necessary to cut off more scrap there than elsewhere to ensure that the remaining plate be solid, and the longitudinal seams being there it is the most important part of the plate. Taking a

Defects in Ends of Plates.

test strip from these parts generally ensures that the ends of the plates are quite equal to the test strip, and that all the defective material has been cut away. The ends of the plates occasionally are spongy or laminated; these defects, however, if they fall within the part from which the test strip is cut, can generally be seen in the fractured part of the strip after it has been pulled asunder.

The test strips from the other plates to be taken from the most convenient part of the plates, according to the usual custom at the works.

The tests of the plates to be witnessed by one of the National Boiler and General Insurance Company's Inspecting Engineers, when those plates found satisfactory by him will be stamped with the Company's stamp, close to the brand. The boiler maker must satisfy himself that each plate is so stamped before using it in the construction of the Boiler.

A list of the plates to be furnished by the maker, giving the consecutive or index number of each, with its breaking stress and elongation, as obtained by testing the strips cut therefrom. The test number and brand to be stamped on each plate, once about the centre and again at one corner, about 1 foot from the adjacent sides. These brands and stamps to be encircled by a ring of white paint, so that they may be easily found.

THICKNESS OF PLATES.

The specified thickness of plates means that the plate should not be less than that thickness.

The thickness is to be measured near the edges of the plates, in view of the fact that it is most important that plates should be up to thickness at the laps and seams.

PLATE DEFECTS.

PLATE DEFECTS.

Below are mentioned some of the defects which are met with from time to time.

Cause of Laminations.

Laminations are usually caused by blow holes in the ingots when cast. When the plate is rolled they extend in the same manner as the plate. If the plate is of great length in proportion to its width the lamination will be of similar form. Thus, in shell

Various Forms of Laminations.

plates, while laminations from blow holes rarely extend more than a few inches across the end, they may be several feet in length. Blow holes sometimes form in the centre of the ingot, the material shrinking away from the centre towards the outside, and the impurities in the steel to some extent collecting there. This class of defect is called "piping," and when it occurs a large lamination, with defective material adjoining, is formed, and is serious. Another class of defect is that caused by small cracks in the sides of the ingots when cast. These fractures are flattened down into the plate, and are usually detected by a faint irregular line running across or along the plate. When this is chipped it is found to extend in a wedge shape to a slight depth into the plate and terminate. Another class of lamination is caused by the rolls folding over the end of the bloom when rolling it into a plate. When finishing the plate this folded portion is rolled closely together, and sometimes is not entirely sheared off. When laminations of this kind occur at the end of shell plates they will generally run across the full width of it, but not extend to any great length.

Pitted Surface.

Another class of defect is caused by the oxide being rolled into it, and causing pittings. These defects, if not disclosed previously, generally show themselves by the scale cracking off when the plates are being bent to form. Foreign material is also sometimes rolled into the plate, causing deep, sharp pittings and embedding grit into the plate.

II.

EXTERNAL SHELLS. RIVETING.

SHELLS.

SHELLS.

It is advised that each ring of the shell should be constructed of one plate, and in Horizontal Boilers the longitudinal seam should be arranged in the steam space and above the flue covers.

In some cases, as in Marine type Boilers, it is of the first importance to avoid ring seams, and it may occasionally be desirable, therefore, in the design of a Boiler of this class, to arrange to have two plates in the ring, and thus have two longitudinal seams and no ring seam.

BENDING OF PLATES.

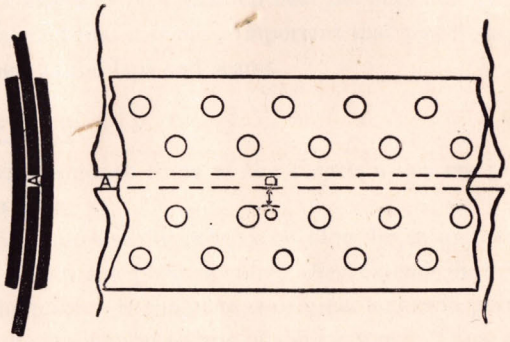
The plates for shells and flues should be bent to true circular cylindrical form by suitable rolls.

Special means should be adopted to bend the ends near the longitudinal seam.

This bending of the ends should never be done by hammering.

LONGITUDINAL SEAMS OF SHELL BUTT JOINTS.

In butt joints, the ends, preferably, should butt close together and ought not to be more than $\frac{1}{8}$ inch apart. (*See* **A** on sketch.) The distances between the edge of the plate and the rivet holes should not be unduly reduced by the gap in the shell plates, or by not placing the butt straps central over the seam. (*See* **C D** on sketch.)



The butt straps, as a rule, to be the same thickness as the shell plates they connect.

The thinning of butt straps for tucking under at the ends, and also of plate corners for tucking under at lap joints, is preferably done by machining, not by heating and hammering.

SHELL CIRCULAR SEAMS.

Care is needed to arrange the lengths of the inner and outer belts of shell plating so that with butt joints the plate ends butt closely without undue gap. The usual practice is to arrange the outer rings just a little short, so that there will be no risk of the plate ends butting before the outer shell ring is tightly gripping the adjacent inner rings at the two circular seams. When the rings are bolted in position and are being drilled it should be noted that there is no clearance between the plates at circumferential laps. Such a fulness in the outer rings cannot afterwards be pressed in by the riveting machine.

Riveting of Ring Seams.

Shell ring seams are generally single riveted. In Lancashire and Cornish Boilers, when the plates are $\frac{9}{16}$ inch thick and over, the ring seams should be double riveted. In such cases it is recommended that single riveting be adopted for the seams of the shell angle ring to the front end and shell and for the seam of the back end to the shell.

RIVET HOLES.

RIVET HOLES.

All rivet holes should be drilled from the solid plate after the plates have been bent to form and are fixed together in the position they will occupy when finally riveted up. After this drilling through the solid has been done the plates should be taken apart, and the rough edges formed in drilling the holes taken off with a countersinking tool, so that the edge of each hole is left with a slight countersink, and no burr remains such as might lie between the plates and prevent a tight joint being made.

Drilling Rivet Holes.

TACKING HOLES.

All tacking holes should be drilled.

RIVET HEADS.

RIVET HEADS.

The rivet heads should be sound, well and fully formed, central on shank, free from lip, and well closed.

Form of Rivet Heads.

Conical Heads.

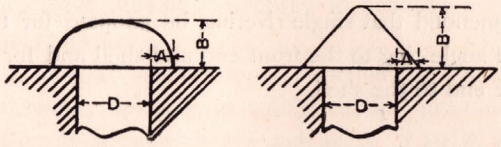
Conical heads are sometimes used where hand riveting is adopted, and where it is not practicable to form a snap head or cup head on the rivet. The form of the conical head is often too flat. This is naturally adopted by the riveters, as it is easier to form a low, flat head than a full-proportioned head, as indicated on the sketch.

It is desirable that the heads should not be less than is indicated in the following :—

Cup Heads or Snap Heads.

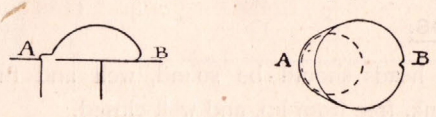
Cup or Snap Head Proportions.—The thickness or height (**B**) of the head should not be less than one-half the diameter (**D**) of the rivet. The lap or shoulder (**A**) of the head should not be less than $\frac{1}{4}$ inch for $\frac{3}{4}$-inch rivets, and $\frac{5}{16}$ inch for $\frac{7}{8}$-inch rivets.

Conical Head Proportions.—The lap or shoulder of the head at each side (**A**) should be about $\frac{1}{4}$ inch for $\frac{3}{4}$-inch, and $\frac{5}{16}$ inch for $\frac{7}{8}$-inch rivets, and so on. The height of the rivet head (**B**) should be not less than three-quarters of the diameter (**D**) of the rivet.



Full and strong heads are particularly necessary where the rivets have to withstand a tensile stress, as in the case of the rivets connecting the gusset angles to the end plates, the shell angles to the end plates, and the fitting blocks to the shell, and in similar positions on Lancashire and Cornish Boilers.

If the cup is a little smaller than the rivet head a slight lip is formed all round ; if this lip is concentric it is satisfactory. If the head is strongly lipped it indicates that too long a rivet has been used. On the other hand, it is to be noted that the attempt to ensure the absence of the slightest lipping often results in too small a head and the cup indenting the plates, which is more



objectionable than lipping. The rivet heads should be central on shank, and not eccentric. The sketch shows a badly eccentric head, which is lipped at **A**, but has not quite filled the cup at **B**, and shows a small crack at **B**. Such heads indicate that the cup forming the head has not been central with the point of the rivet when the pressure was applied. This may be caused by the holes not being fair, causing the rivet to be inserted at an angle, the surfaces of the plates not being at right angles to the machine, or the cups not being quite opposite to each other, owing to the machine not being in order.

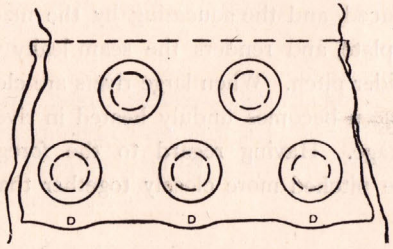
RIVETS WITH COUNTER-SUNK HEADS.

Countersunk heads are frequently used for hand-riveted work in new Boilers, and also in repairs. They are suitable for parts where it is desired to preserve a smooth surface, and where the rivets are in shear; when, however, rivets are in tension, cup-headed rivets should preferably be used.

MACHINE RIVETING.

Excessive Riveting Pressure

The pressures used in the riveting machine should be sufficient to close the rivets properly and securely without bulging the plate. Sometimes, where too high a pressure is utilised, the edges of the lap are bulged, owing to the excessive pressure round the rivet, as indicated on the sketch **D D D**.



The bulging is not only at the edge of the plates, but is also between the rivet holes, and in the case of longitudinal seams, as the plate cannot extend in the longitudinal direction, it tends to arch between the rivet holes.

When this bulging occurs at the ring seams it stretches the plates between the rivet holes and thereby increases the circum-

ference, and at the same time the diameter of the shell at the seam, and this increased diameter can be seen by applying a straight edge along the shell plate.

When the rivet heads are being formed by the riveting machine the pressure should be kept on by the machine for a short period; that is to say, the cup head should not be at once withdrawn from the rivet as soon as the head has been formed, otherwise the spring of the plate may stretch the hot rivet and lead to leaky work. The pressure should be kept on sufficiently long for the rivet to cool down a little, so as to acquire strength to withstand the spring of the plates and to retain them in close contact.

DESIGN OF RIVETING.

DESIGN OF RIVETING.

The design of riveting must be such as not to subject any part to unduly high stress, and also must be arranged so as to give a watertight joint and to permit satisfactory work in the riveting.

Too wide a pitch of rivets allows plates to spring up between the rivets, and prevents proper caulking and fullering.

Too close a pitch causes undue squeezing of plate near rivet, and may result in springing off and leakage.

When rivets are spaced too close together the nett plate section is unduly reduced, and the squeezing by the machine sometimes distorts the plate, and renders the seam leaky which would be tight with a wider pitch. When large rivets are close together also, the plate between becomes unduly heated in riveting, which also tends to leakage. Having regard to the foregoing, the rivets should not be pitched more closely together than is given in the table below.

RIVETING.	MINIMUM PITCH.	
Finished size of Rivet.	Single Riveting.	Double Riveting.
$\frac{3}{4}$ inch.	$1\frac{3}{4}$ inch.	2 inches.
$\frac{13}{16}$,,	2 inches.	$2\frac{1}{4}$,,
$\frac{7}{8}$,,	$2\frac{1}{8}$,,	$2\frac{1}{2}$,,
$\frac{15}{16}$,,	$2\frac{1}{4}$,,	$2\frac{3}{4}$,,

Generally the diameter of the rivets should be fixed according to the thickness of plates which are to be riveted together, and should be somewhat as given below. (The rivet in no case should be less in diameter than thickness of plate.)

DIAMETER OF RIVETS.

Thickness of Plate.	Diameter of Rivet. Finished Size.
$\frac{3}{8}$ inch.	$\frac{3}{4}$ inch.
$\frac{7}{16}$ "	$\frac{13}{16}$ "
$\frac{1}{2}$ "	$\frac{13}{16}$ "
$\frac{9}{16}$ "	$\frac{13}{16}$ "
$\frac{5}{8}$ "	$\frac{7}{8}$ "
$\frac{11}{16}$ "	$\frac{7}{8}$ "
$\frac{3}{4}$ "	$\frac{7}{8}$ "
$\frac{13}{16}$ "	$\frac{15}{16}$ "
$\frac{7}{8}$ "	$\frac{15}{16}$ "
$\frac{15}{16}$ "	$\frac{15}{16}$ "

DETERMINATION OF THICKNESS OF SHELL.

The following rules apply to shells up to 35 feet in length and 9 feet in diameter :—

Tensile Stress on Plates.

THICKNESS OF SHELL PLATE.—The thickness of mild steel plate is to be determined by the ordinary formula (*see pages* 17 to 19), so that the calculated tensile stress on the shell plate shall not exceed the following :—

INTERNALLY-FIRED BOILERS WITH DOUBLE-RIVETED BUTT JOINTS AT LONGITUDINAL SEAMS.—The stress should be between 11,000 pounds and 12,000 pounds, and should not exceed 12,000 pounds per square inch.

INTERNALLY-FIRED BOILERS WITH LAP JOINTS AT LONGITUDINAL SEAMS.—The stress should be between 10,500 pounds and 11,500 pounds, and should not exceed 11,500 pounds per square inch.

Externally-fired Boilers, with Double-riveted Butt Joints at Longitudinal Seams.—Length of Boiler not exceeding 20 feet. The stress should be between 9000 pounds and 10,000 pounds, and should not exceed 10,000 pounds per square inch. In all Externally-fired Boilers the longitudinal seams should not be in contact with the furnace gases.

Externally-fired Boilers, with Lap Joints.—Length of Boiler not exceeding 20 feet. The stress should be between 8500 pounds and 9500 pounds, and should not exceed 9500 pounds per square inch.

If the length exceeds 20 feet, 500 pounds must be deducted from the stress allowable for each additional 10 feet of length.

The longitudinal seams should not be in contact with the furnace gases.

AIR RECEIVERS.

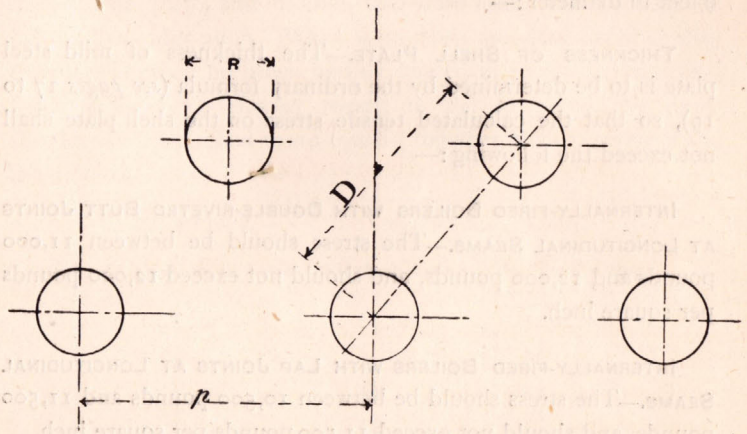
Where vessels such as Air Receivers are not subject to external or internal corrosion, tensile stress allowable about 13,000 pounds., not to exceed 13,500 pounds.

DOUBLE ZIG-ZAG RIVETING.

In the design of double zig-zag riveting, the diagonal pitch (**D**) not to be less than

$$= \cdot 65\, p + \cdot 35\, R.$$

Where p = pitch of rivets R = diameter of rivets.



DETERMINATION OF THICKNESS OF SHELL.

Shearing Stress on Rivets.

In the design of riveting for the longitudinal and ring seams of Steam Boilers the shearing stress on the rivets is not so important as the tensile stress on the plate. As regards rivets in shear, when they once are riveted up and fill the hole they are not subjected to the weakening effect of corrosion, while the shell is liable to be weakened by corrosion and other influences; hence, in design, the calculated shearing stress on the rivets may be higher than the calculated tensile stress on the plates.

Generally the calculated shearing stress should not exceed 15,000 pounds per square inch.

The sketches (pages 21 to 24) show suitable arrangements of riveting which give satisfactory results:—

RIVETING.

The formulæ and symbols used in calculating the examples of riveting (pages 21 to 24) are as follow:—

p = Pitch, inches.

R = Diameter of rivet, inches.

A = Area of one rivet, square inches.

D = Diameter of Boiler, inches.

P = Pressure, pounds per square inch.

T = Thickness of plate, inches.

N = Number of rivets in one pitch.

= 2 for double riveting.

Tension on strip of external shell 1 inch wide $= P \times \frac{D}{2}$

Tension on strip one pitch wide $= P \times \frac{D}{2} \times p$

Tensile stress on plate, pounds per square inch $= \frac{P \times \frac{D}{2} \times p}{(p - R) \times T}$ . . . . . (1)

Let $K = \frac{\text{percentage strength of joint}}{100}$

Call K for plate between rivet holes $= K_t$

„ shearing on the rivets $= K_s$

„ combined shear and tension $= K_{st}$

and evidently equation (1) may be written :—

$$\text{Tensile stress, pounds per square inch} = \frac{P \times \frac{D}{2} \times p}{K \times p \times T}$$

$$= \frac{P \times D}{2 \times K \times T} \quad . \quad . \quad . \quad (2)$$

To obtain the tensile stress on the plate K_t must be used in this equation for K.

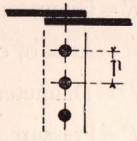
To obtain the shearing stress on the rivets K_s must be used in this equation for K.

Single-riveted lap joint:—

$$K_t = \frac{(p - R) \times T}{p \times T}$$

$$= \frac{p - R}{p}$$

$$K_s = \frac{A}{p \times T}$$

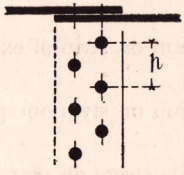


Double-riveted lap joint:—

$$K_t = \frac{(p - R) \times T}{p \times T}$$

$$= \frac{p - R}{p}$$

$$K_s = \frac{2\,A}{p \times T}$$

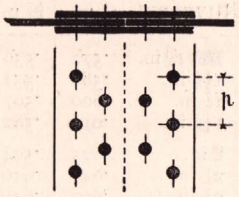


Double-riveted butt joints.

For rivets in double shear total effective area to be taken for each rivet = 1·75 A.

$$K_t = \frac{(p - R) \times T}{p \times T} = \frac{p - R}{p}$$

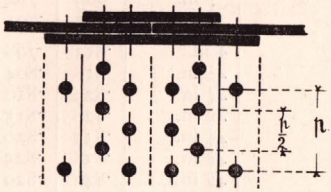
$$K_s = \frac{2 \times 1{\cdot}75 \times A}{p \times T} = \frac{3{\cdot}5\ A}{p \times T}$$



Treble-riveted butt joint, with narrow outside straps and wide inner straps.

For outer row :—

$$K_t = \frac{p - R \times T}{p \times T} = \frac{p - R}{p}$$

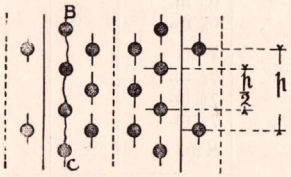


Supposing plate intact and all rivets sheared

$$K_s = \frac{A + 4 \times 1{\cdot}75\ A}{p \times T} = \frac{8 \times A}{p \times T}$$

K_{st}—Supposing shear outer row rivets and tear plate along **B C**

$$K_{st} = \frac{A + 2\left(\frac{p}{2} - R\right) T}{p \times T} = \frac{A + (p - 2\ R)\ T}{p \times T}$$

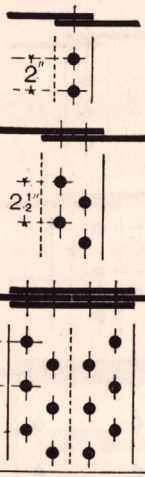


The table which follows gives $K_t = \frac{\text{percentage strength of plate}}{100}$ for various pitches of riveting and thicknesses of plate :—

Pitch of Riveting.	Diameter of Rivets.						
	$\frac{3}{4}$ in.	$\frac{13}{16}$ in.	$\frac{7}{8}$ in.	$\frac{15}{16}$ in.	1 in.	$1\frac{1}{16}$ in.	$1\frac{1}{8}$ in.
[illegible] $1\frac{3}{4}$ in.	·572	·536					
$1\frac{13}{16}$ in.	·586	·551					
$1\frac{7}{8}$ in.	·600	·567					
$1\frac{15}{16}$ in.	·613	·581					
2 in.	·625	·594	·562				
$2\frac{1}{8}$ in.	·647	·616	·588				
$2\frac{1}{4}$ in.	·667	·639	·611				
$2\frac{3}{8}$ in.	·684	·658	·631				
$2\frac{1}{2}$ in.	·700	·675	·650	·625			
$2\frac{5}{8}$ in.	·714	·691	·667	·643			
$2\frac{3}{4}$ in.	·727	·704	·682	·659			
$2\frac{7}{8}$ in.	·739	·717	·696	·673			
3 in.	·750	·729	·708	·687			
$3\frac{1}{8}$ in.	·760	·740	·720	·700			
$3\frac{1}{4}$ in.	·769	·750	·730	·712			
$3\frac{3}{8}$ in.	·777	·758	·740	·723			
$3\frac{1}{2}$ in.	·785	·767	·750	·732			
$3\frac{5}{8}$ in.	·793	·776	·759	·741			
$3\frac{3}{4}$ in.	·800	·784	·767	·750			
$3\frac{7}{8}$ in.	·807	·791	·774	·758			
4 in.	·813	·798	·781	·765			
$4\frac{1}{8}$ in.	·818	·804	·788	·772			
$4\frac{1}{4}$ in.	·823	·810	·794	·779			
$4\frac{3}{8}$ in.	·828	·815	·800	·785			
$4\frac{1}{2}$ in.	·833	·820	·806	·791	·778		
$4\frac{5}{8}$ in.	·838	·825	·811	·797			
$4\frac{3}{4}$ in.	·842	·829	·816	·802	·790		
$4\frac{7}{8}$ in.	·846	·833	·821	·807			
5 in.	·850	·837	·825	·812	·800	·790	
$5\frac{1}{4}$ in.	·857	·844	·833	·820	·810		
$5\frac{1}{2}$ in.	·864	·851	·841	·828	·818	·806	
$5\frac{3}{4}$ in.	·870	·858	·848	·836	·825		
6 in.	·875	·864	·855	·844	·833	·822	·813
$6\frac{1}{4}$ in.				·851	·840	·830	
$6\frac{1}{2}$ in.				·857	·846	·837	
$6\frac{3}{4}$ in.					·852	·844	
7 in.					·857	·850	·840
$7\frac{1}{4}$ in.						·855	
$7\frac{1}{2}$ in.						·860	
$7\frac{3}{4}$ in.						·864	
8 in.						·867	·860

The NATIONAL Boiler and General Insurance Co. Limited, Manchester.

$\frac{3}{8}$" PLATE
WITH $\frac{3}{4}$" RIVETS
FINISHED SIZE



SINGLE LAP

$K_t = \cdot62, K_s = \cdot59$

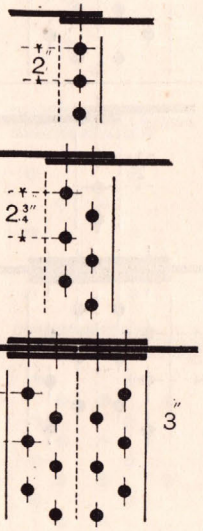
DOUBLE LAP

$K_t = \cdot7, K_s = \cdot94$

DOUBLE BUTT

$K_t \cdot72, K_s = 1\cdot5$

$\frac{3}{8}$" PLATE
WITH $\frac{13}{16}$" RIVETS
FINISHED SIZE



SINGLE LAP

$K_t = \cdot59, K_s = \cdot69$

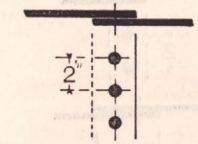
DOUBLE LAP

$K_t = \cdot7, K_s = 1\cdot0$

DOUBLE BUTT

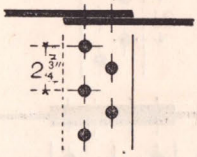
$K_t = \cdot73, K_s = 1\cdot6$

$\frac{7}{16}$" PLATE
WITH $\frac{13}{16}$" RIVETS
FINISHED SIZE



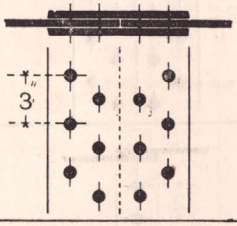
SINGLE LAP

$K_t = \cdot59, K_s = \cdot59$



DOUBLE LAP

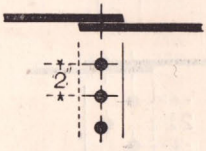
$K_t = \cdot7, K_s = \cdot86$



DOUBLE BUTT

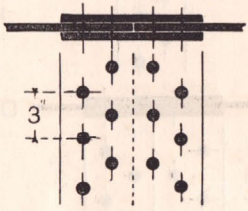
$K_t = \cdot73, K_s = 1\cdot38$

$\frac{1}{2}$" PLATE
WITH $\frac{13}{16}$" RIVETS
FINISHED SIZE



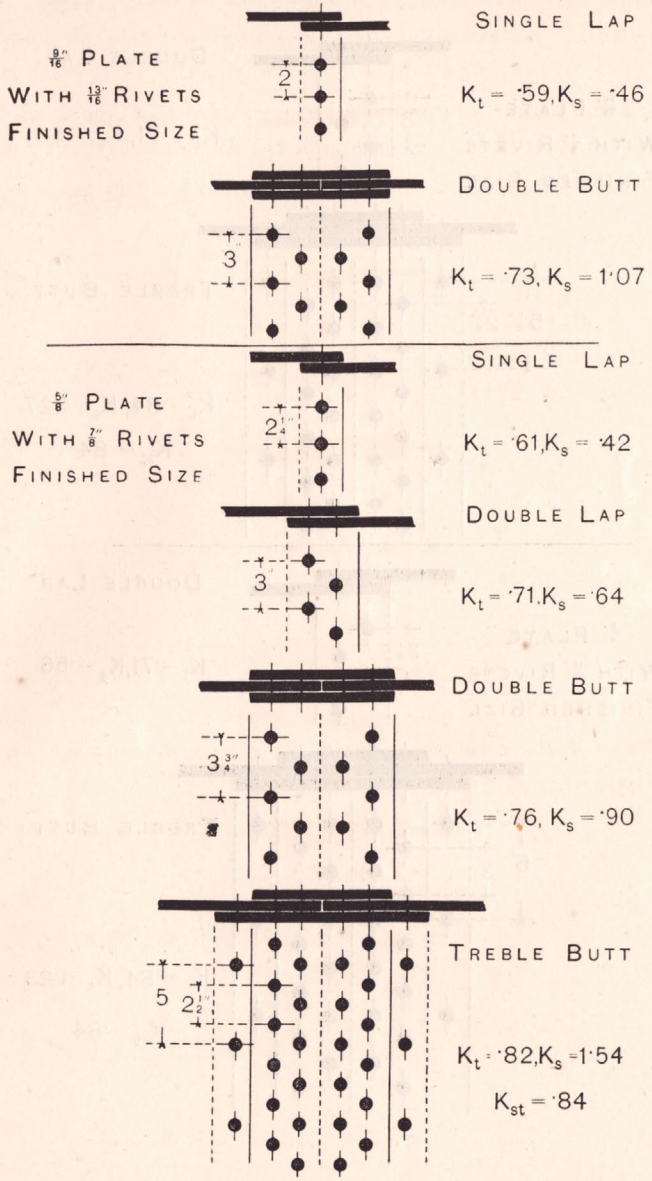
SINGLE LAP

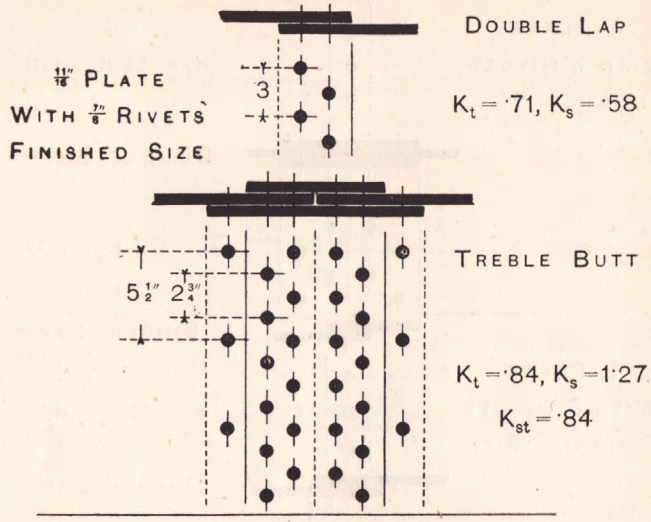
$K_t = \cdot59\ K_s = \cdot52$

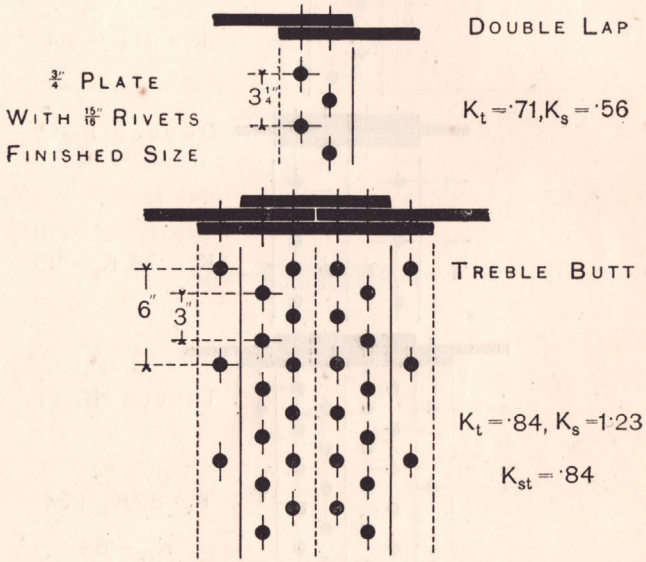


DOUBLE BUTT

$K_t = \cdot73, K_s = 1\cdot21$


SINGLE LAP
9/16" PLATE
WITH 13/16" RIVETS
FINISHED SIZE
2
K_t = ·59, K_s = ·46
DOUBLE BUTT
3
K_t = ·73, K_s = 1·07
SINGLE LAP
5/8" PLATE
WITH 7/8" RIVETS
FINISHED SIZE
2 1/4"
K_t = ·61, K_s = ·42
DOUBLE LAP
3
K_t = ·71, K_s = ·64
DOUBLE BUTT
3 3/4"
K_t = ·76, K_s = ·90
TREBLE BUTT
5
2 1/2"
K_t = ·82, K_s = 1·54
K_st = ·84


11/16" PLATE
WITH 7/8" RIVETS
FINISHED SIZE
3
DOUBLE LAP
Kt = ·71, Ks = ·58
5 1/2" 2 3/4"
TREBLE BUTT
Kt = ·84, Ks = 1·27
Kst = ·84


3/4" PLATE
WITH 15/16" RIVETS
FINISHED SIZE
3 1/4"
DOUBLE LAP
Kt = ·71, Ks = ·56
6"
3"
TREBLE BUTT
Kt = ·84, Ks = 1·23
Kst = ·84

Copper Plates in cylindrical shells subject to internal pressure, putting the plates in tension, not exposed to heating by furnace or flue gases and containing water or saturated steam. COPPER PLATES IN TENSION.

In design, as a rule, arrange that maximum tensile stress does not exceed 3000 pounds per square inch.

Copper plates should not be used for superheated steam.

III.

MANHOLES AND MUDHOLES. BOLTS.

PROVISION FOR INTERNAL AND THOROUGH INSPECTION.

Every Boiler should be provided with suitable manholes, mudholes, and sightholes. These are required to enable the Boiler to be cleaned, and also are essential for proper inspection, as required by the Factory and Workshop Act and similar laws. Suitable provision is arranged as a matter of course in large Boilers, such as Lancashire or Water-tube Boilers, but in small Boilers, such as small Locomotives, Motor Wagons, Traction Engines, etc., the provisions for internal examination and cleaning are frequently very limited.

When the Boiler is large enough, it is always best to provide openings sufficiently large to allow the Boiler to be entered.

When, however, the Boiler is too small for this, special attention should be given to providing suitable sightholes and mudholes for cleaning and inspection, and also, if possible, at least one opening large enough to allow a man's head to pass through.

MANHOLES AND MUDHOLES.

MANHOLES AND MUDHOLES.

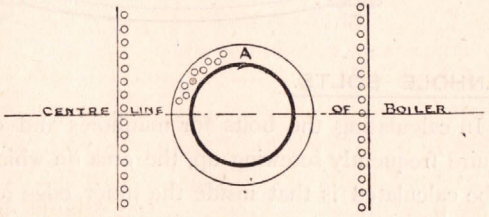
It is advised that all manhole and mudhole mouthpieces and covers be made of mild steel, pressed into form.

Oval manhole mouthpieces, with their internal covers, are preferably made of pressed steel, the faces being machined, the clearance between the spigot and the mouthpiece not exceeding $\frac{1}{16}$ inch each side, or a total of $\frac{1}{8}$ inch, to avoid risk of jointing being blown out.

The bolts of such internal covers should be substantially riveted in or attached by strong nuts. OVAL MANHOLES.

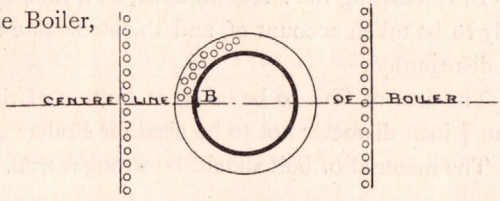
Oval manholes in the cylindrical shell should be placed with their shorter axes along the Boiler.

Welded circular manhole frames to be fixed so that the weld may lie across the Boiler, thus—



WELDED MANHOLE MOUTH-PIECES.

and not along the Boiler, thus—



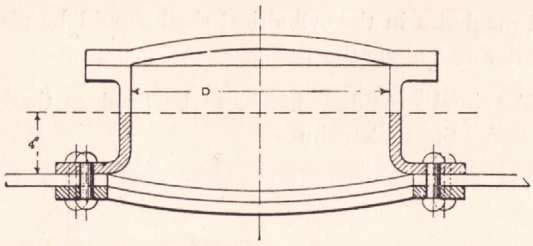
In Boilers for pressures above 100 pounds the manhole mouthpieces, whether raised or otherwise, should be weldless.

For pressures above 100 pounds per square inch, as a rule, a strengthening ring, extending beyond the flange of the mouthpiece, is advisable. WELDLESS MOUTH-PIECES.

Such rings and mouthpieces should be arranged with the idea of gathering up the lines of force in the adjacent plate and distributing them on the other side of the opening.

The sectional area, excluding parts cut out by rivets, provided by way of compensation within 4 inches of the shell, should exceed the area of plate cut away by the manhole opening by at least 10 per cent of the area cut away.

The shaded section shows the part to be included in the 4-inch calculation.



MANHOLE BOLTS.

MANHOLE BOLTS.

In calculating the bolts for manholes and other parts which require frequently screwing up, the area on which the pressure is to be calculated is that inside the inner edge of the joint. (*See* **D** in above sketch.)

In calculating the stress on bolts, as a rule, the steam pressure only to be taken account of, and the stress due to screwing up to be disregarded.

The area of bolt to be taken at bottom of thread. Bolts less than $\frac{3}{4}$ inch diameter not to be used for Boilers of ordinary size.

The material of bolt should be wrought iron.

MANHOLE BOLTS OF BOILERS WHICH ARE SCREWED UP AT INTERVALS OF A FEW WEEKS.

Material of bolt, wrought iron.

The maximum stresses to be allowed to be as on table below :—

Diameter of Bolt.	Allowable Stresses.
$\frac{3}{4}$ inch	3000 pounds per square inch.
$\frac{7}{8}$,, ...	4000 ,, ,,
1 ,, ...	5000 ,, ,,
$1\frac{1}{8}$,, ...	5500 ,, ,,
$1\frac{1}{4}$,, ...	6000 ,, ,,
$1\frac{3}{8}$,, ...	6500 ,, ,,
$1\frac{1}{2}$,, ...	7000 ,, ,,

BOLTS FOR MANHOLES OF RAG BOILERS, KIERS, OR SIMILAR VESSELS WHICH ARE SCREWED UP DAILY OR OFTENER.

For such bolts, as a rule, the highest quality of Yorkshire iron, or an equivalent quality, is advised. The maximum stresses allowable to be as under:—

Diameter of Bolt.	Allowable Stresses.
$\frac{3}{4}$ inch	2500 pounds per square inch.
$\frac{7}{8}$,, ...	3500 ,, ,,
1 ,, ...	4500 ,, ,,
$1\frac{1}{8}$,, ...	5000 ,, ,,
$1\frac{1}{4}$,, ...	5500 ,, ,,
$1\frac{3}{8}$,, ...	6000 ,, ,,
$1\frac{1}{2}$,, ...	6500 ,, ,,

MANHOLE BOLTS AND COVERS FOR LANCASHIRE BOILERS.*

The Boilermakers' Committee recommend:—"The covers of round raised pattern, wrought iron or steel manholes 16 inches or 17 inches internal diameter, be of dished steel plates 1 inch thick, with 16 bolts 1 inch diameter for working pressures up to 120 pounds inclusive, and $1\frac{1}{4}$ inches thick with 16 bolts $1\frac{1}{8}$ inches diameter for pressures over 120 pounds up to 200 pounds inclusive."

SIZES OF OVAL MANHOLE FOR FRONT END PLATE.

The Boilermakers' Committee recommend:—"That the following be the standard size of manhole for front end of Boilers below the flues:—15 inches by 11 inches for Boilers 6 feet 6 inches to 8 feet diameter both inclusive, and 16 inches by 12 inches for Boilers above 8 feet diameter."

* The Boilermakers' Committee of the Manchester District Engineering Trades Employers Association.

STAND PIPES OR FITTING BLOCKS.

STAND PIPES OR FITTING BLOCKS.

It is advised that the stand pipes and blocks for the fittings should be of mild steel, pressed or welded, and worked into form.

There are several methods in use of constructing the blocks. In some cases the flanges are welded on—*i.e.*, by a weld at the corners—and in others the block is formed by bending round a properly-formed angle piece. The blocks which are made by bending round and welding up a suitably formed angle are generally better and more reliable than those which are made by welding the flanges on.

In all cases of wrought blocks the thickness of the flanges requires special attention, also the bending of the corners. Blocks which show surface fractures at the corners should usually be rejected, as these indicate that the metal is brittle or has been bent at an improper heat, and has probably been seriously damaged.

There should be no sharp corners anywhere in the block. These are liable to lead to grooving and fracture.

Weldless mild steel blocks, pressed from the solid, are preferred for pressures over 100 pounds.

RIVETING OF STAND PIPES, BLOCKS, OR PADS TO BOILER.

The stand pipes or other blocks or pads require to be carefully bedded to the Boiler, and the riveting suitably pitched to ensure a tight joint.

Common causes of leakage in working at these parts are: Stand pipe or block not properly bedded before riveting; flange of stand pipe too thin; rivets too far apart from each other, as in some cases where only four rivets are used for the attachment of the feed valve pad to the front end.

LEVEL OF FACES OF BLOCKS.

Care is required to ensure that stand pipes or blocks are truly centred in the Boiler, and that their machined faces are set correctly, so that the fittings will be in their right level and position when the Boiler is set.

STAND PIPES AND BLOCKS FOR FITTINGS.

For Lancashire Boiler and other large Boilers weldless pressed steel stand pipes are preferred for the attachment of the fittings.

The stand pipes should have a moderate taper only, so that an unduly large hole in the steel is not necessitated.

As a rule, the diameter of the hole in the shell should not exceed the smallest diameter of the stand pipe by more than $2\frac{1}{2}$ inches.

For a stand pipe of this class no reinforcing ring is required, provided that the stand pipe and flange net section, within 4 inches of the external surface of the Boiler, exceeds by not less than 10 per cent the area of the shell cut away by the opening and rivet holes.

The determination of the compensating area to be made as indicated on page 27 for manholes. If the stand pipe, as first designed, is not adequate to give the required compensation, then compensation may be obtained by a thicker flange, by extending the flange and double riveting it to the shell, or by adopting a compensating ring in addition to the stand pipe. As a rule, it is preferred to single rivet the stand pipe to the shell rather than to double rivet it, unless considerations of strength render the latter course desirable.

Strengthening Rings for Stand Pipes.

Any strengthening rings for stand pipes for fittings on the top of a Horizontal Boiler to be internal.

Strengthening rings for stand pipes on the bottom of a Horizontal Boiler to be external.

It is desirable, except for very thick plates, that the flange of the stand pipe which is riveted to the shell should not be thinner than the shell.

STEAM DOMES OR DRUMS CONNECTED BY NECKS.

When steam domes or drums are connected to shells, generally one large neck is preferable to two, so as to avoid straining due to differences of expansion.

IV.

FLAT AND CAMBERED ENDS.

FLAT ENDS.

FLAT ENDS OF STEAM DRUMS AND SIMILAR VESSELS OF MILD STEEL.

For the ends of cylindrical vessels the flat unstayed form is undesirable, and should only be adopted where cambered or stayed ends cannot be utilised.

If flat ends used, to be designed as follows :—

To be *Mild Steel.*

Flanged by machine and annealed.

Rivet holes drilled.

Thickness not less than cylindrical shell to which it is riveted.

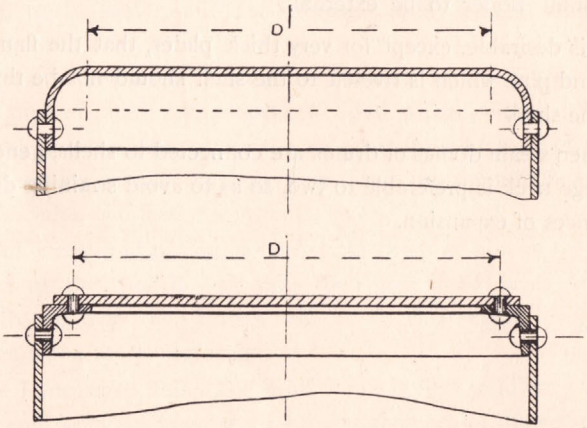
Radius of corner to be as large as can be arranged, and not less than four times thickness of plate.

No allowance to be made for any riveted blocks or other stiffening on the flat crown.

Rule to apply to diameters up to $D = 3$ feet 6 inches, and thickness up to $t = 16$, that is, 1 inch thick.

t = thickness in sixteenths of an inch.

p = pressure, pounds per square inch.



In flanged ends (*see* sketch) take the circle of diameter **D**, which includes the flat part of crown. This diameter = distance apart of centres of radii of corners.

In case of crown with *angle ring attachment* take circle to centre of rivets.

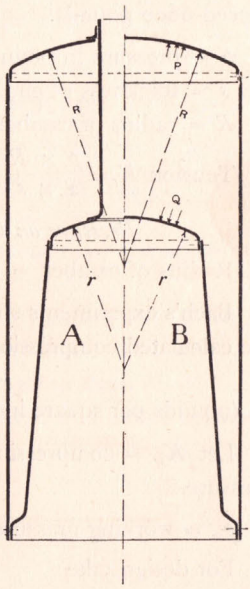
Area of circle (diameter D) $= \cdot 7854\ D^2$
$= S.$

Then for fixing pressure $= p$
use formula

$$p = \frac{K \times t^2}{S} \quad K \text{ being taken} = 300 \quad . \quad . \quad (1)$$

CAMBERED ENDS.*

Material: Mild Steel (*Flanging Quality*).—24 to 28 tons tensile strength; elongation not less than 20 per cent in 10 inches.



CAMBERED ENDS OR DISHED ENDS OF STEAM BOILERS.

RADIUS OF CAMBER.

Recommended to be equal to about two-thirds the diameter to a maximum equal to the diameter of the cylindrical part to which end is attached.

RADIUS OF CORNER OF FLANGE.

Recommended to be one-eighth of the radius of the camber, and in no case less than two inches. It is important that this radius should be as large as practicable.

End cambered or dished by machine, suitable dies being used, so that the camber is truly spherical.

No hammering or flogging to be allowed.

Anneal after all flanging and fitting done.

* NOTE.—Hemispherical ends are preferred. Cambered or dished ends should only be used where hemispherical ends are impracticable.

Rivet holes drilled.

Rule to apply up to a maximum diameter of cylindrical part of 4 feet 6 inches.

The thickness of end should never be less than the shell to which it is attached, so as to ensure a good joint.

In Vertical Boilers with cambered ends to the shell and furnace, as shown on diagram (half-section **A**), the ends should be calculated on the above basis without taking account of support of chimney tube, as indicated on half-section **B**.

Pressure on Concave Side of end as at P.

Cambered End Pressure on Concave Side.

Tension K_t = calculated tensile stress allowable, should not exceed 6000 pounds.

p = pressure in pounds per square inch.

s = thickness of end in inches.

R = radius of camber in inches.

$$\text{Tension } K_t = \frac{p \times R}{2 \times s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Pressure on Convex Side of End as at Q.

Pressure on Convex Side of End.

Radius of camber = r.

Bach's experiments show that

the calculated compression at bulging = K_o is given by

$$K_o \text{ (pounds per square inch)} = 36979 - 1636 \sqrt{\frac{r}{s}} \quad . \quad . \quad . \quad (2)$$

Let K_w = compression, pounds per square inch, at working pressure.

p_w = working pressure, pounds per square inch.

For design take

working compression, pounds per square inch, $K_w = \frac{1}{5} K_o$

$$= 7396 - 327 \sqrt{\frac{r}{s}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

K_w should not exceed 5000 pounds; then, provided $\frac{r}{s}$ does not exceed 53, a compression of 5000 pounds per square inch may be used in design.

If $\frac{r}{s}$ exceeds 53, the K_w, or compression allowable, should be found by equation (3). (*See* note *.)

Working compression, pounds per square inch = K_w

$$= \frac{p_w \times r}{2s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

These rules give minimum thickness, and the thickness so obtained may be increased, if necessary, when special wear is likely to occur, such as in the case of the furnace crown of Vertical Boilers.

As the tests of cambered ends show that the greatest stress exists near to the bends of the flanges, the radius of the corner should be kept as large as convenient, and in the case of large ends it will often be desirable to double rivet the end to the cylindrical part.

CAMBERED ENDS OF LANCASHIRE BOILERS.

CAMBERED ENDS OF LANCASHIRE BOILERS.

Dished or cambered ends of Lancashire Boilers with pressure on concave side of plate as at **P** (*see* figure, page 33), differ from the cambered end of a plain cylindrical shell, because—

a. The end is relieved of the pressure over the areas cut out by the tubes.

b. The tubes support the end—the support thus given is greater with flue tubes constructed with ordinary flanged seams than with corrugated flues.

Ends of this class are generally cambered to a radius between one diameter of the Boiler and one and a half diameters of the Boiler.

To determine thickness of ends use equation (1) page 34.

$$K_t = \frac{p \times R}{2 \times s}$$

* NOTE.—Maximum $K_w = 5000 = 7396 - 327\sqrt{\frac{r}{s}}$

$$\frac{2396}{327} = \sqrt{\frac{r}{s}}$$

$$\frac{r}{s} = 53{\cdot}6$$

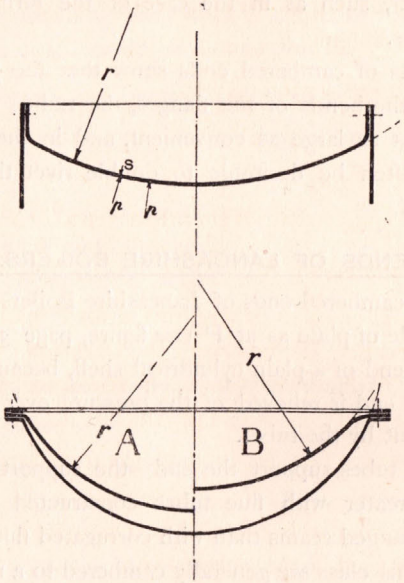
(say 53) when the compression is equal to 5000 pounds, the maximum allowable. As 5000 pounds should not be exceeded, the principle stated above, in other words, is to use equation (3) when the resulting compression does not exceed 5000, but when the application of equation (3) would give K_w above 5000, then $K_w = 5000$ to be used.

When the flue tubes are constructed with ordinary flanged seams $K_t = 11{,}000$.

When the flue tubes are corrugated tubes $K_t = 9000$.

Copper Cambered Ends.

Copper cambered ends, as for brewing pans, jacketed pans. Well-hammered copper (not annealed). Spherical dished ends. Subject to pressure on convex side of end (*see p p* diagram); such, for instance, as brewing pans, confectionery boiling pans.



The spherical curve should come well up to the abutment, as at **A**, and not a wide section subject to bending, as at **B**.

Then the calculated compression in pounds per square inch at which bulging will commence was shown by Bach's experiments to be $= K_o$ in equation (5), which follows :—

r = radius of camber, inches.

s = thickness of copper, inches.

p_o = pressure in pounds per square inch at which bulging took place.

$$K_o = 36268 - 1706 \sqrt{\frac{r}{s}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$\frac{p_o \times r}{2s} = K_o \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (6)$$

Let K_w = compression, pounds per square inch, at working pressure.

p_w = working pressure pounds per square inch.

In designing new vessels make working pressure = $\frac{1}{6}$ calculated bulging pressure.

And the calculated working compression $K_w = \frac{1}{6}$ of the compression at bulging K_o.

$$K_w = .\frac{1}{6}\left(36{,}268 - 1706\sqrt{\frac{r}{s}}\right) \quad \ldots \ldots \ldots \quad (7)$$

$$= 6044 - 284\sqrt{\frac{r}{s}}$$

Do not use compression higher than 3000. (*See* note *)

Working compression $K_w = \frac{p \times r}{2s}$

These are allowances where the dished part starts from a good abutment. If, however, the form at the abutment is such as to reduce the power of resistance of the dished end, as at **B**, then the pressure allowable must be reduced.

An example of a pan designed on the foregoing lines is given on page 38.

In the foregoing it is assumed that saturated and not superheated steam is used in the jacketed pan.

---

* NOTE.—

Maximum $K_w = 3000 = 6044 - 284\sqrt{\frac{r}{s}}$

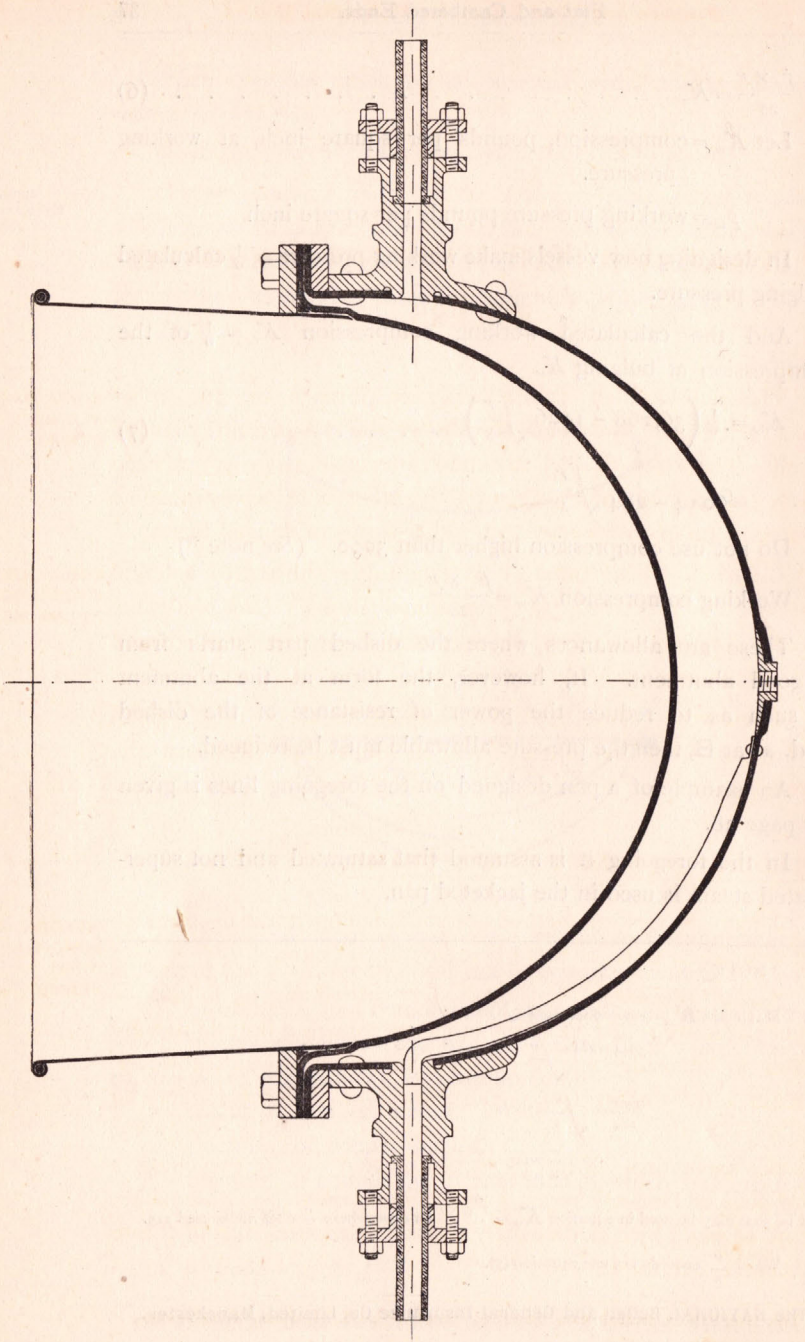
$$3044 = 284\sqrt{\frac{r}{s}}$$

$$\frac{3044}{284} = \sqrt{\frac{r}{s}} = 10{\cdot}72$$

$$\frac{r}{s} = 115$$

that is, 3000 may be used in equation $K_w = \frac{p_w \times r}{2s}$ for cases where $\frac{r}{s}$ does not exceed 115.

When $\frac{r}{s}$ exceeds 115 use equation (7).



COPPER JACKETED PAN.

V.

FURNACE AND FLUE TUBES FOR HORIZONTAL BOILERS.

FORM OF FLANGED SEAMS.

The notes which follow apply to furnace or flue tubes constructed with the seam known as the "Adamson" flanged seam. The strength and thickness (*see* table, page 45) advised apply only to this form of seam; the other notes apply in some degree to tubes constructed with other forms of flanged seam.

CIRCULAR CYLINDRICAL FURNACES.

General.—In all furnaces or flue tubes subject to external pressure avoid riveted lap longitudinal seams. Use preferably *welded longitudinal seams*, or else, though not so good, riveted butt joints *with double straps.*

Circularity.—True form is assumed in these rules; small deviations from the circle have an important effect on the strength.

WELDING OF LONGITUDINAL SEAMS.

WELDING OF LONGITUDINAL SEAMS.

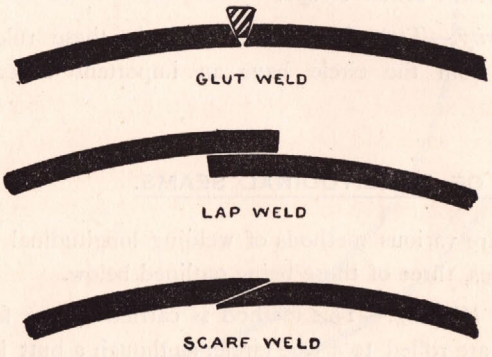
There are various methods of welding longitudinal seams in the flue tubes, three of these being outlined below.

"*Glut*" *Welding.*—This method is carried out as follows:—The plates are rolled to a true circle, as though a butt joint were about to be made; the chamfered ends of the plates, however, remain a little apart (*see* sketch), straps or chains being placed round the tubes to prevent spreading when welding proceeds. At the same time as the plates are heated on the two edges, a long strip of iron, say about $\frac{1}{2}$ inch wide, $\frac{3}{4}$ inch deep to 2 inches deep, or say $\frac{1}{2}$ inch in diameter, is also heated, and is welded into the space left for it.

This method usually results in a sound and tight job, and little fullering is afterwards necessary.

Lap Welding.—The plates are rolled until the ends of the plates lap well over each other. (*See* sketch.) The plates are then heated and welded up. There are no thin edges exposed to the fire, and consequently the risk of burning is said to be reduced. Frequently firms tack the laps together by, say, three rivets, to prevent spreading, and straps round the tube are also used for the same purpose. Sometimes this method is modified by reducing the lap to $\frac{1}{4}$ inch or $\frac{1}{2}$ inch, and omitting the rivets. Sometimes the inside plate is scarfed, the outer being as on sketch.

"*Scarf*" *Welding.*—In this case the edges of the plates are bevelled with a long scarf, and the plates are rolled until the edges just come well over each other. (*See* sketch.) The plates are then welded one to the other, being prevented from spreading by hoops round each tube or one or two rivets. This method is



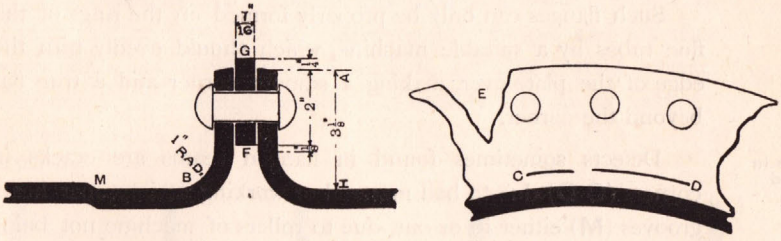
very frequently adopted, and usually gives good results, although some firms say that the thin edges of the scarf are liable to overheating if the greatest care be not exercised.

In all these methods of welding, a length of from 6 inches to 1 foot is finished at a time if the best machine be used, but with hand welding usually less is done at each stage.

After the weld has been completed there ought to be no openings or apparent fractures on either side of the weld, and the weld throughout the length of the ring should be even and smooth. After the rings have been welded they are usually, after truing up, put on the flanging machine and the flanges formed. The stretching of the metal at the flanges very frequently, in the operation of flanging, causes the weld to open across the flange, as at E. (*See* sketch.) This part is usually made good by means of the welding in of a triangular piece of plate, and, providing this is done in a careful manner, the work is satisfactory.

FLAWS IN WELDS.

Circularity of Welded Rings.



A better plan, however, is as follows:—When welding the ring longitudinally, instead of lap welding the ends of the joint, make a glut weld at each end by welding in a lump of soft iron; this lump becomes spread over the joint, strengthening it at these parts and increasing the thickness. The end at the weld is then knocked out, bell-mouthing it a little and preparing it for flanging, and also testing the weld. Rings welded in this way will seldom crack at the joint in flanging.

The welds of the rings of the flue tubes should be at the bottom, and break joint about 12 inches.

FLANGED SEAMS.

The sketch shows the proportions advised for ordinary Adamson flanged seams. The bend of the flange (B) should be struck with an inner radius of not less than 1 inch. The

FLANGED SEAMS.

Proportions.

flat of the flange should finish to at least $2\frac{1}{8}$ inches broad, so that when the outer edge of the flange has been turned there will be a total flange (A H) of about $3\frac{1}{8}$ inches, reckoning from the edge to the outside of the ring body. The caulking ring (F G) will be about $2\frac{1}{4}$ inches wide, and the inner edge should be of sufficient diameter to leave $\frac{1}{8}$ inch of the flat of the flange uncovered. This flat facilitates caulking of the ring, avoids the caulking ring having to be spread in order to fill the space between the two flanges, and gives a little elasticity to the joint. It is also desirable that the outer edges (A) of the two flanges should be caulked against the caulking ring.

Caulking Ring

Such flanges can only be properly formed on the rings of the flue tubes by a suitable machine, which should evenly turn the edge of the plate over, making a smooth corner and a true flat beyond the corner.

Defects sometimes found in flanged seams are cracks in corners (C D) due to bad material or working at improper heat; grooves (M) either in or out, due to rollers of machine not being properly set; undue and unequal thinning of the flanges.

Defects in Flanged Seams.

CROSS TUBES.

Cross tubes are inadvisable in tubes 2 feet 6 inches diameter and under, as they block up the tube so much as to seriously interfere with cleaning and inspection.

CROSS TUBES.

When cross tubes are inserted it is preferable that they should be riveted to the plate by flanges formed on the cross tubes. Welding of the cross tubes into the flue tube is a common practice, and although it does not appear practicable to form even and thoroughly-sound welds at each end of the cross tube, the welds which are made seem generally to give satisfaction in working. The cross tubes require carefully cutting to length, so that they may not unduly draw in or bulge out the rings of the flue tube.

The welds of cross tubes should be at the back, the principle being that the welds should not be exposed to the direct impact of the flame. Position of Welds.

CONNECTION OF TUBES TO ENDS.

CONNECTION OF TUBES TO ENDS.

There are a number of plans in ordinary use for connecting flue tubes and similar parts to the end plates.

A.—With angle ring. Advantages: Simple and easy to construct; easy to adjust length of tube. Disadvantages: Rivets exposed to flame; also difficulty of accurate fit of tube in angle.

B.—Tube flanged; usually this tube ring made $\frac{1}{16}$ inch to $\frac{1}{8}$ inch thicker than others. Advantages: Simple and efficient when well constructed; requires care to get ring correct length. Disadvantage: Leakage troubles occasionally with high pressure; grooving in bend of flange.

C.—Similar to **B**, but tippet piece at back of flange; ring same thickness as others; common plan. Disadvantages: In drilling holes burrs get forced between tippet piece and flange, and lead to leakage; grooving under tippet piece cannot be detected.

D.—End flanged in; a good joint if well constructed. Disadvantages: Requires careful fitting of tube in end; can only be hand riveted, and rivets exposed to flame; not so good for front ends as **E.**

E.—End flanged out; a good plan for high pressures; can be machine riveted. Disadvantage: Requires very accurate fitting of tube in end, and unsuitable for some forms of mechanical stoker.

HORIZONTAL CIRCULAR FURNACES AND FLUE TUBES.

HORIZONTAL CIRCULAR FURNACES AND FLUE TUBES.

Design of mild steel tubes with flanged seams.

Make the front end rings not less than $\frac{1}{16}$ inch to $\frac{1}{8}$ inch thicker than the others, and in order to distribute any bending, and avoid local distress at the flanges, these rings, as a rule, should be the same thickness as the end plate.

Usually make the back end rings about $\frac{1}{16}$ inch to $\frac{1}{8}$ inch thicker than others.

Design of Flue Tube with Flanged Seams.

Determine thickness of other rings by table, page 45.

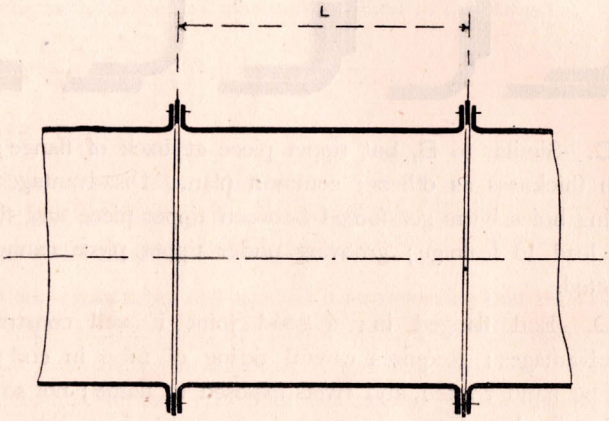
Arrange position of flanged seams to be—

Not opposite shell ring seams ;

Not directly over furnace firebridge.

For stiffness in case of overheating do not arrange flanges more than 3 feet 6 inches apart as a rule.

Longitudinal seams, as a rule, to be welded, and to break joint on bottom.



In Lancashire Boilers cone down last ring but one, make diameter of last ring = general diameter – 6 inches.

In Cornish Boilers cone down last ring to diameter = general diameter – 6 inches.

For furnace and flue tubes made with flanged seams, of the form shown at page 41, the proportions given on the following table are advised:—

TABLE OF THICKNESSES AND DISTANCE APART OF FLANGED SEAMS.

By "Length" in column below is meant the distance apart of the centres of consecutive flanged seams. (*See* L on diagram page 44.)		Pressures, pounds per square inch.						
Diam.		80	100	120	140	160	180	200
ft. in.		ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
2 0	Length	3 5	3 0	2 8	2 5	2 3	2 1	2 0
	Thickness...	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{15}{32}$
2 3	Length	3 5	3 0	2 8	2 5	2 3	2 1	2 0
	Thickness..	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{15}{32}$	$\frac{1}{2}$
2 6	Length	3 5	3 0	2 8	2 5	2 3	2 1	2 0
	Thickness..	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{17}{32}$
2 9	Length	3 5	3 0	2 8	2 5	2 3	2 1	2 0
	Thickness...	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{17}{32}$	$\frac{9}{16}$
3 0	Length	3 5	3 0	2 8	2 5	2 3	2 1	2 0
	Thickness...	$\frac{13}{32}$	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{17}{32}$	$\frac{9}{16}$	$\frac{19}{32}$
3 3	Length	3 5	3 0	2 8	2 5	2 3	2 1	2 0
	Thickness...	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{15}{32}$	$\frac{17}{32}$	$\frac{9}{16}$	$\frac{19}{32}$	$\frac{5}{8}$
3 6	Length	3 5	3 0	2 8	2 5	2 3	2 1	2 0
	Thickness...	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{17}{32}$	$\frac{19}{32}$	$\frac{5}{8}$	$\frac{21}{32}$

The foregoing table applies only to the diameters given therein, and not exceeding 3 feet 6 inches diameter.

NOTE.—Mainly from consideration of wearing qualities, the minimum thickness in the above table has been increased from $\frac{3}{8}$ inch to $\frac{13}{32}$ inch.

VI.

FLAT SURFACE RULES.

STAYING OF FLAT SURFACES.

STAYING OF FLAT SURFACES.

In applying the following rules, regard must be paid to the suitability of the flat plate to transmit the stress to the stay which supports it, and also to the suitability of the stay for withstanding the stress so transmitted to it by the plate.

The first depends on the stiffness of the plate, and is dealt with in the section on the strength of plates.

The second depends on the strength of the stay, and is dealt with for screw and bolt stays below.

BOLT STAYS.

BOLT STAYS.

As a rule, it is advisable that bolt stays should be solid, and not welded, and in designing stays such forms should be adopted, wherever practicable, as can be made from the solid without welding.

In the case of all screw stays, such as firebox stays of Locomotive-type Boilers, it is advantageous to turn the stay down to a diameter not exceeding that of the bottom of the thread.

Tensile Stress Allowable, Calculated on Area of Stay at Bottom of Thread, or on Least Cross-Sectional Area of Stay.

STRENGTH OF STAYS.

Screw Stays of Locomotive Firebox and Similar Parts, with Ends Riveted over to Form Substantial Heads.

Material.		Tensile Stress.
Stay.	Plate.	
Iron (quality as below)	Iron	8000 pounds per square inch
Iron	Copper	6000 ,, ,, ,,
Copper	Copper	5500 ,, ,, ,,
Steel	Steel	8000 ,, ,, ,,

RIVETED STAYS.

Bolt Stays, with Nuts, or Screw Stays, with Nuts. (Not to be welded).*

Material.	Tensile Stress.
Stay.	
Iron (quality as below)	9000 pounds per square inch
Steel	9000 ,, ,, ,,

NUTTED STAYS.

It is advised that, if iron be used for screw stays or bolt stays, it should be Farnley, Low Moor, or equal quality.

In bolt stays, as a rule, it is better that the body of the stay should be rather less in diameter than the bottom of the thread.

APPLICATION OF RULES FOR STAYING OF FLAT SURFACES.

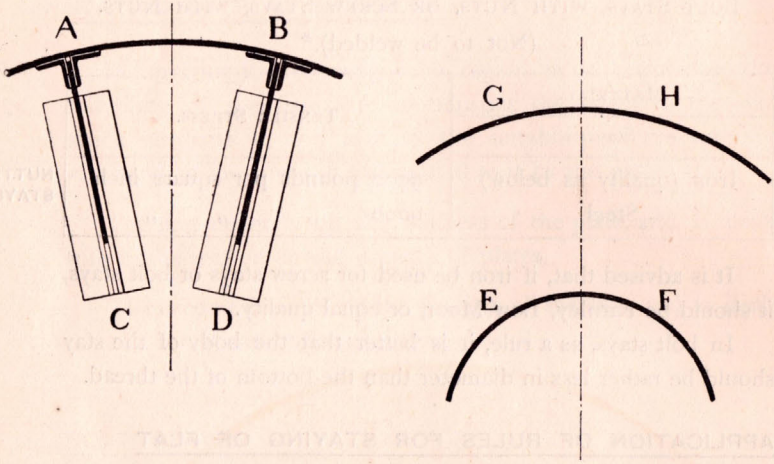
STRENGTH OF PLATE.

Flat surfaces are stayed and supported in so many different ways that the question of which rule should be applied requires careful consideration.

It is to be noted that the areas of flat plates enclosed by the supporting stays, or supporting lines, as in gussets, vary considerably in shape. For instance, where bolt stays are

* It is recommended that bolt stays should not be welded, and therefore the stresses allowable for welded stays are omitted.

pitched in a rectangular equidistant manner, the spaces towards the middle will be rectangular in form ; those towards the edges of the plate surface may be surfaces which are bounded by straight lines in one direction and curves in another. Spaces between gusset stays, as in Lancashire and Cornish Boilers, also vary very much in the shape of the boundaries and the nature of their support. For instance, the portion **A B** of the end plate which is supported by the seam to the shell angle, or which lies near the bend of flange in the case of back-end plates, is supported practically rigidly, in the ordinary sense, all round, while that portion of the plate which lies near the gusset stays, **A C**, **B D**, is supported by a stay which is in itself moderately elastic.



The support afforded by the boundaries **A C**, **B D** of such a flat surface will not be so rigid as is the support of the boundary **A B**, or as indicated in the end of a Cornish Boiler **E F**, **G H**, where the shell and the tube act as longitudinal stays in supporting this surface.

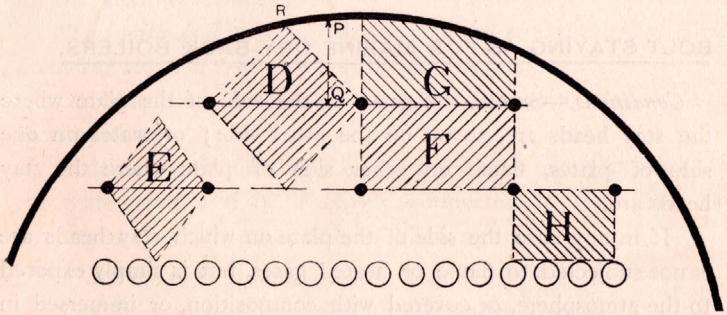
Hence, when deciding what particular constant of the gusset rules to apply to an end plate supported by gussets, the general guiding principle to be observed is :—That constant should be

adopted which is stated in the rule to be applied to bounding stays, or bounding edges of plates which have similar degrees of elasticity, to those which exist in the particular case in question.

Again, in the case of flat surfaces supported by bolt stays, it is to be remembered that these rules for bolt stays are derived directly from experimental results, and these experimental results relate in all cases to the areas enclosed between bolts spaced in a rectangular manner.

Most of these experiments have been conducted with rectangular boxes through which a number of stays pass, and the stayed boxes generally failed by the giving way of the stays or plate toward the centre of the box. This is only natural, because it is obvious that the riveted edges of such boxes afford a much more complete support to the plate adjacent to them than do the bolt stays. The consequence is, that, unless the bolts are placed at a greater distance than the ordinary pitch from the riveted edges of the boxes, the plate between the riveted edge of the box and the bolt stays is more strongly supported than the plate between the rectangular bolt stays.

This fact has an important bearing on the application of the rectangular bolt-staying rule to many end plates. Thus, for instance, in the figure below, in considering the pitch to be



employed in utilising the ordinary stay rule for the outer parts of the plate, the vertical distance (**P Q**) shown must not be taken as the pitch of the staying, but it should be assumed

that the staying will be about as effective as if at the edge of the end plate (*see* diagram) there were another stay **R**, and the areas taken to calculate the strength of the plate are those marked **D**, **E**, and **F**, and not those which are marked **G**, **H**, and which at a first glance might seem to be the most suitable.

For central parts of plate take area **F**.

STIFFNESS OF PLATE.

RULES RELATING TO THE STIFFNESS OF THE PLATE.

Flat ends, stayed by gussets or screw or bolt stays.

C = Constant for each description of staying.

T = Thickness of plate in sixteenths of an inch.

S = Area supported by one stay.

With *Bolt Stays* S usually = vertical pitch × horizontal pitch. (*See* page 49.)

With *Gusset Stays* S = area of the largest circle which can be inscribed within an unstayed area of the end, the circumference of the circle passing through the centre of rivets of gussets or other support.

(Also, as to S generally, *see* notes, pages 48 and 53.)

p = suitable pressure, pounds per square inch.

$$p = \frac{C \times T^2}{S} \quad . \quad . \quad .

Regularly-spaced bolt staying or screw staying, as in Locomotive or Marine type Boilers.

Stays riveted over to form substantial heads. Stays with Riveted Heads.

Steel plates ... ... $C = 110$

Copper plates ... ... $C = 60$

Stays screwed through plates and nutted externally. Nutted Stays.

Steel plates ... ... $C = 130$

Stays passing through the plate (which not screwed), with nuts each side with or without an ordinary washer.

Steel plates ... ... $C = 140$

Stays with double nuts and substantial large washers riveted to the plate. Nutted Stays with Large Washers.

Thickness of washer not less than half thickness of plate, and diameter of washer about half pitch of stays if thickness of washer equals thickness of plate, to two-thirds pitch of stays if thickness of washer equals half thickness of plate. Approximate Limits of Size of Washers.

$$C = 250$$

This applies particularly to the staying of the upper parts of the ends above the smoke tubes of Boilers of the Marine type.

Sometimes small ordinary washers are interposed between the nuts and the large riveted washers. No special allowance made for the small washers.

Stays with double nuts. No washers, but riveted doubling plate extending over the area supported by the longitudinal stays. Nutted Stays with Doubling Plates.

Doubling plate preferably external, and covering the whole of the stayed area.

If this riveted doubling plate is as thick, or nearly as thick, as the plate itself, and is effectively connected to that plate by riveting at the outer edge, use the following rule :—

T_1 = Thickness of outer plate in sixteenths of an inch.

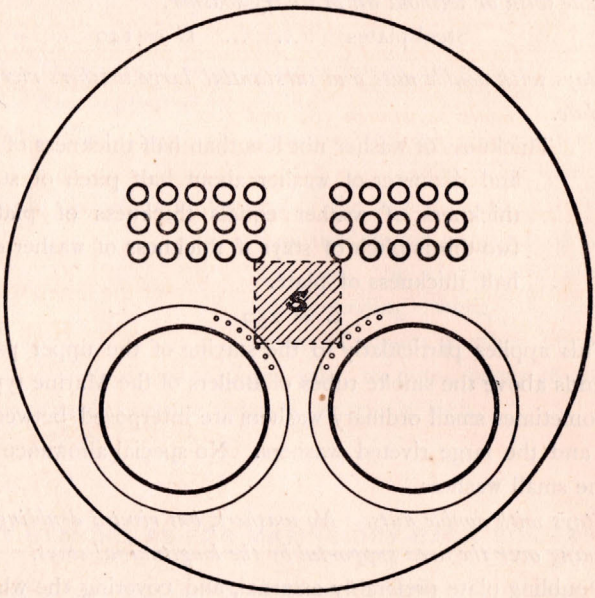
T_2 = Thickness of inner plate in sixteenths of an inch.

$$p = \frac{C \times (T_1^2 + T_2^2)}{S} \quad \ldots \ldots \ldots \ldots \quad (2)$$

$C = 140$.

This applies particularly to the staying of the upper parts of the ends of Marine type Boilers.

Cases will occur in which the doubling plate is hardly large enough to cover the whole of the stayed area, as is assumed in the foregoing. In such cases the pressure should he fixed between that given for stays with double nuts with large washers and that given for stays with large doubling plates, as described above.



Marine type Dry-back Boilers.

For steel Boilers of the Marine Dry-back type the space below the smoke tubes and above the flue tubes at the back bend usually determines the thickness of the end plate. (*See* sketch.)

In formula :—

$$p = \frac{C \times T^2}{S}$$

take $C = 220$.

GUSSET STAYS OF FLAT ENDS, AS IN LANCASHIRE AND CORNISH BOILERS.

GUSSET STAYS.

In formula $p = \frac{C \times T^2}{S}$, previously given, S is taken as the area of the largest circle which can be inscribed in an unstayed space of the ends, the circumference of the circle touching the centre line of the nearest stay rivets, as **P**, **Q**, **T**, **R**. (*See* diagrams below.)

The gusset staying should be so spaced that the value of the constant C may be as follows :—

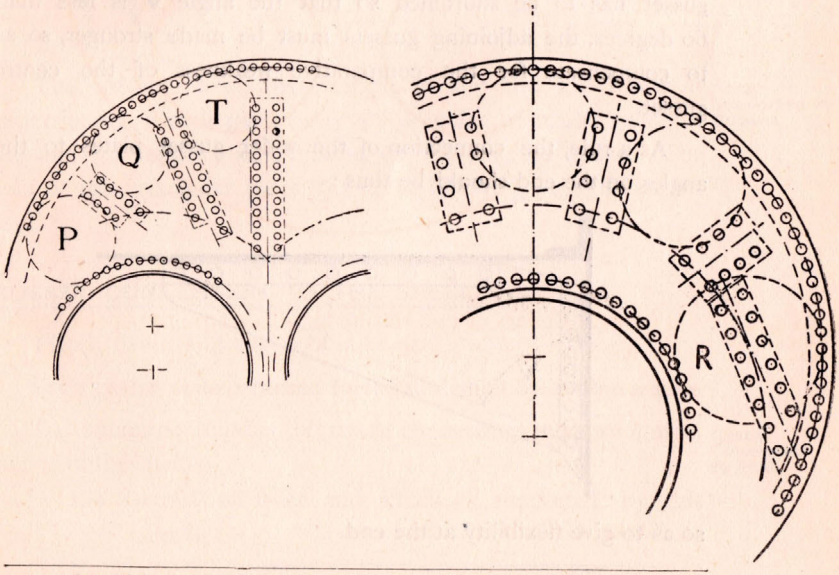
Spacing of Gussets.

Steel Gusset Plates.—Double angles to shell and end. In Lancashire and Cornish Boilers, for spaces between two gussets such as **Q**, **T**, *C should be about 220, and should be between 190 and 250.*

Lancashire Boilers.—Spaces outside wing gussets, such as **P**, *C should be about 250 and should not exceed 300.*

Cornish Boilers.—Spaces outside wing gussets as at **R**, *C should not exceed 280.*

If when using the equation $p = \frac{C \times T^2}{S}$ for the outer space in a Cornish Boiler it is difficult to arrange the space **R** so that C



shall not exceed 280, then the gusset should be so arranged that in no case should S for the space R be so large as to necessitate a greater constant than 400, and in all cases where the constant exceeds 280 a steel stiffener is to be used. This steel stiffener may be so arranged that it passes nearer than the breathing circle to the sides of the flue tube, as indicated on the diagram. It is to be remembered that the sides of the flue tube do not move so much, owing to expansion and contraction, as the top, and therefore the same breathing space is not essential at these parts.

If the constant corresponding to the space R should exceed 400, then some rearrangement of the gusset stays should, as a rule, be made so as to bring it within this limit.

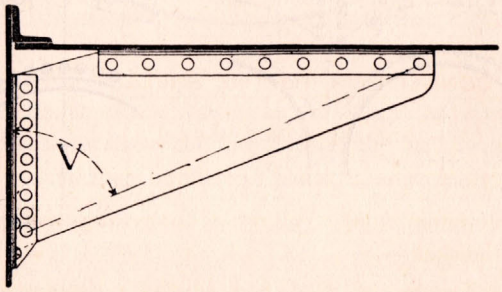
GUSSET PLATE AND RIVETS.

Gusset Plate.

The angle V, between the end plate and the line joining bottom rivet to end and the furthest rivet to shell, should not be less than 60 degrees.

If, owing to the nearness of a manhole to the end plate any gusset has to be shortened so that the angle V is less than 60 degrees, the adjoining gussets must be made stronger, so as to compensate for the comparative weakness of the centre gusset.

As a rule, the connection of the wider gusset plates to the angles on the end should be thus :—



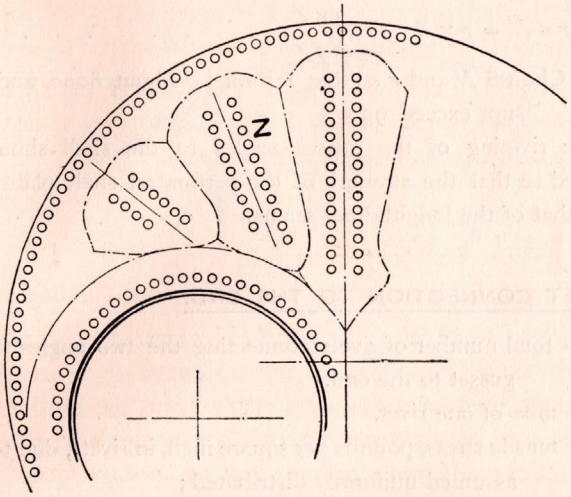
so as to give flexibility at the end.

CALCULATION OF STRESS ON GUSSET RIVETS.

AREA SUPPORTED BY GUSSET.

Stress on Gusset Rivets.

For the purpose of this calculation, it is to be assumed, as far as the gusset staying is concerned, that each part of the end is supported by the nearest rivet connecting the gusset stays to the end or the tube or shell to the end.



Area Supported by Gussets.

The application of this principle to any end will result in apportioning to each gusset stay an irregular triangular piece of the end for support, as indicated on the diagram above. The area of this triangular piece will in the following notes be known as *Z*.

GUSSET CONNECTION TO THE SHELL.

Plates, rivets, and angles of mild steel.

Each gusset plate attached to end and shell by double angles.

Gusset Connection to Shell.

To determine number of rivets connecting the two gusset angles to the shell—

Let Z = area of front end which is supported by this gusset.

p = pressure, pounds per square inch.

N = number of rivets connecting the angles of this gusset to shell = 2 × number connecting one of gusset angles to shell.

r = area of one rivet.

f = shearing stress in one rivet, pounds per square inch;

then

$$f \times r \times N = p \times Z \quad f = \frac{p \times Z}{N \times r} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Choose N and r so that f shall be about 8000, and shall not exceed 9500.

The riveting of the gusset angles to the shell should be designed so that the strength of the perforated shell plate is not below that of the longitudinal seams.

GUSSET CONNECTION TO THE END.

Gusset Connection to End.

n = total number of rivets connecting the two angles for one gusset to the end.

r = area of one rivet.

t = tensile stress, pounds per square inch, in rivets, due to load assumed uniformly distributed;

then

$$t \times r \times n = p \times Z \quad t = \frac{p \times Z}{r \times n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Make t about 6000, and not exceeding 8000, and choose n and r accordingly.

It may be necessary in some cases to make angles long, putting rivets a little further apart. In others large angles may be necessary in order to obtain the requisite rivet area.

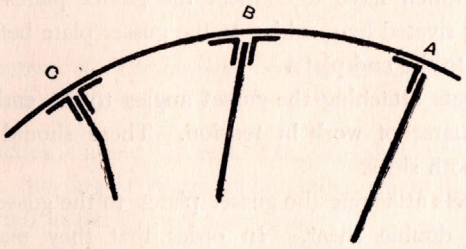
In view of the heavy stress which falls on the toe rivets connecting the gusset angles to the ends of Lancashire Boilers, it is usual in high-pressure Boilers to make these rivets $\frac{1}{16}$ inch greater in diameter than the other rivets at this part.

Gusset Stays and Angles.

The arrangement of the gusset stays in Boilers varies, of course, according to the design. Sometimes these gusset stays are arranged radially; sometimes only the vertical stays point

to the centre of the Boiler, the others being tangential to some circle described round the centre of the Boiler.

When the gusset stays are not radial the gusset plate will not be normal to the shell plate, and therefore, if the angles are to lie close against the gusset plate, as they should do, they must be set according to the angle of the plate. Thus **A** indicates the gusset



plates set radially, the angles being in their ordinary condition; **B** shows a gusset stay which is not radial, and in order to bed truly the angles require setting properly on a block before they are fixed in the Boiler; **C** shows a gusset stay, the stay not being radial to the centre of the Boiler, but the angles have not been set, and consequently the stay plate itself is cranked or bent. This is very undesirable. All gusset plates ought to be straight.

Rivet holes in gusset angles should all be drilled, not punched.

A common mode of attaching gusset stays is as follows:—

Fixing of Gusset Stays in position.

The gusset angles, after being properly drilled, are fixed on the end plates, and riveted thereto in many cases by machine, a suitable plate being temporarily fixed between the angles. The gusset angle rivet holes in the shell are first of all drilled, and the angles, after being duly set to their proper angle, are marked off for drilling. When this marking off is being done a plate of the same thickness as the gusset plate ought to be closely bolted between the gusset angles. The gusset angles are afterwards riveted in position on the shell, and the gusset plate then placed

Grip of Gusset Angle.

between the angles on the shell and those on the end, and marked off for drilling. The gusset angles on the shell and those on the end ought to grip the plate so as to hold it firmly. Want of grip is liable to lead to looseness of the stays and slackness of rivets, and to necessitate the replacement of the rivets in the course of a year or two.

In some cases a variation of the foregoing procedure is adopted, the angles which have to connect the gusset plates to the end plate being riveted by machine to the gusset plate before they are connected to the end plate.

The rivets attaching the gusset angles to the end are in the ordinary course of work in tension. These should have good heads on both sides.

Gusset Riveting.

The rivets attaching the gusset plates to the gusset angles are usually in double shear. In order that they may be more effective in this respect, it is desirable that they should fill the holes and have good heads. The size of the heads, however, is not so important as in the case where the rivets are in tension, and in the parts where the gusset stays are close together at the lower parts on the front end plate it is difficult to form full heads, The gusset rivets, however, should be closed to fill the holes and also have heads sufficient to hold the plates firmly together. otherwise the rivets are liable to work loose after a short period of working.

VII.

LANCASHIRE AND CORNISH BOILERS.

SIZES OF LANCASHIRE AND CORNISH BOILERS.

The diameters of Lancashire Boilers usually lie between 8 feet 6 inches and 6 feet 6 inches.

8 feet 6 inches is about the maximum, and this is fixed by the fact that it is the largest diameter of Boiler which can usually be carried by rail in the United Kingdom.

6 feet 6 inches is about the minimum diameter of Lancashire Boilers. It is not advisable to adopt the Lancashire type for smaller diameters, as the spaces for internal cleaning, and also the diameter of the furnaces, become too small for practical purposes.

30 feet is about the maximum length usually adopted.

Standard sizes are:—

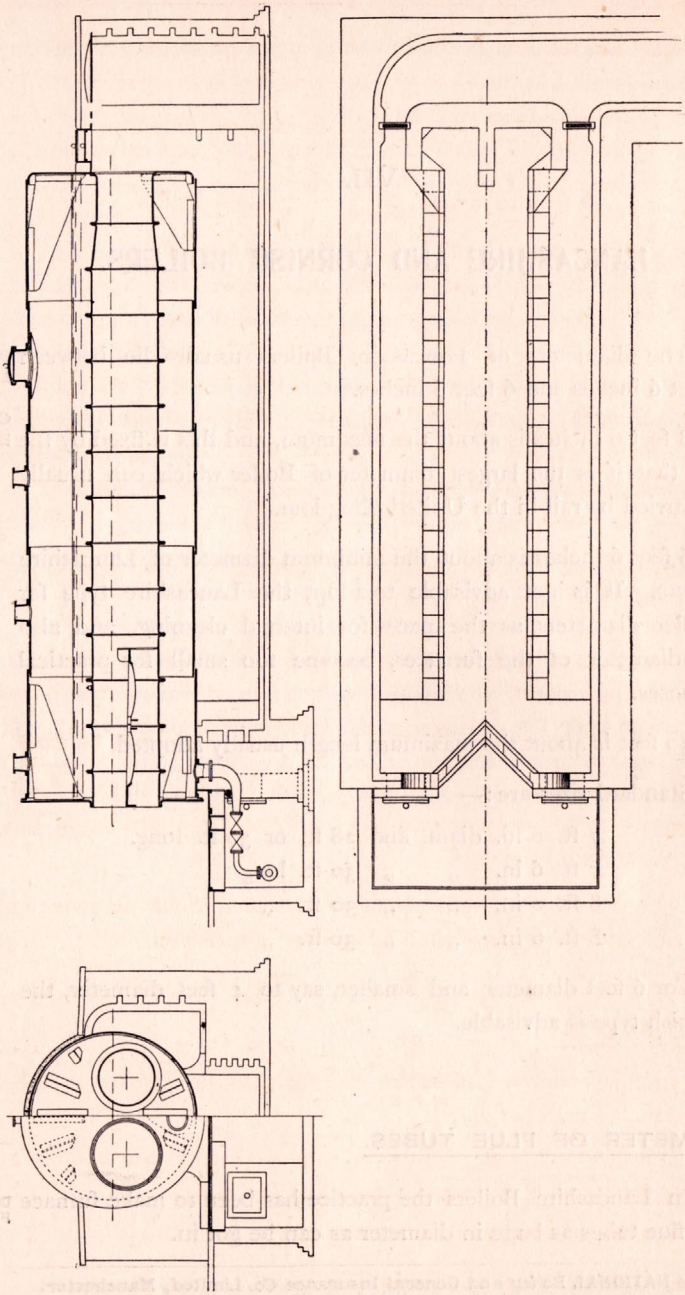
7 ft. 0 in.	diam.	and	28 ft. or 30 ft.	long.
7 ft 6 in.	,,	,,	30 ft.	long.
8 ft. 0 in.	,,	,,	30 ft.	,,
8 ft. 6 in.	,,	,,	30 ft.	,,

For 6 feet diameter and smaller, say to 4 feet diameter, the Cornish type is advisable.

DIAMETER OF FLUE TUBES.

Diameter of Flue Tube.

In Lancashire Boilers the practice has been to make furnace and flue tubes as large in diameter as can be got in.



LANCASHIRE BOILER.

This has resulted in the following approximate proportions :—

Lancashire Boilers.

External Diameter of Flue and Furnace Tubes.	Diameter of Shell.
2 feet 10 inches.	7 feet 0 inches.
3 ,, 1 ,,	7 ,, 6 ,,
3 ,, 3 ,,	8 ,, 0 ,,
3 ,, 5 ,,	8 ,, 6 ,,

If the feed water be sedimentary, so that internal cleaning will require special attention, reducing the diameter of the tubes will give increased space for this purpose, but by increasing the relative water contents will render the Boiler more sluggish in getting up steam, and also give it a greater reserve.

Cornish Boilers.

In Cornish Boilers the relation between the diameter of the shell and the diameter of the tube is :—

$$\left.\begin{array}{c}\text{External diameter}\\ \text{of furnace and}\\ \text{flue tube}\end{array}\right\} \begin{array}{c}\text{is}\\ \text{usually}\\ \text{made}\end{array} = \left\{\begin{array}{l}\dfrac{\text{diameter of shell}}{2} + 1 \text{ inch.}\\ \text{to}\\ \dfrac{\text{diameter of shell}}{2} + 4 \text{ inches.}\end{array}\right.$$

In Lancashire and Cornish Boilers the first shell ring is generally made an outer one, to give room for expansion of the end plate round furnaces.

The last ring is usually arranged so that the outside diameter of back end plate flange shall be an even dimension.

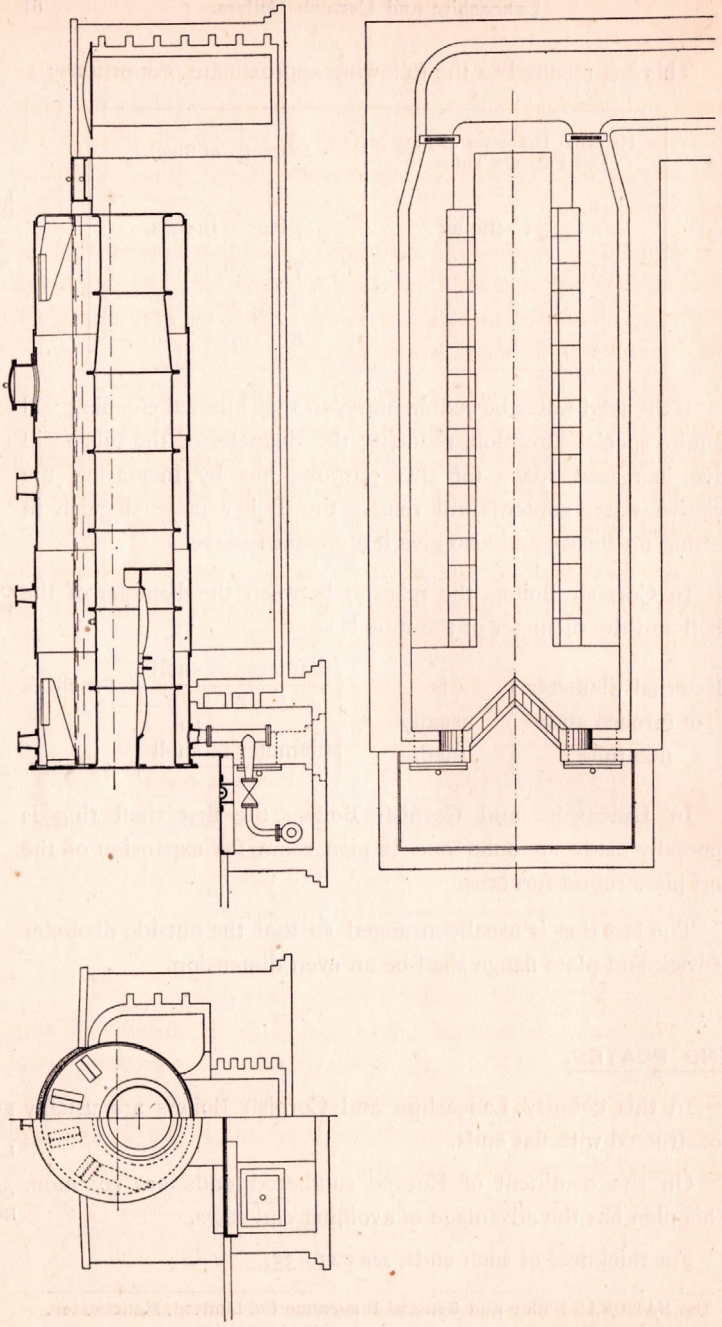
END PLATES.

END PLATES OF LANCASHIRE AND CORNISH BOILERS.

In this country Lancashire and Cornish Boilers are usually constructed with flat ends.

On the continent of Europe cambered ends are common. This plan has the advantage of avoiding end stays.

For thickness of such ends, *see* page 35.



CORNISH BOILER.

Cambered or dished ends require special dies for pressing to form and flanging, and as the ends are more rigid than flat ends the flue tubes should for some part of their length be constructed of corrugated tubes, so as to accommodate longitudinal expansion and contraction.

Flatness of End.

Flatness of Ends.—Where it is intended that the end shall be flat, care should be taken that the flat is true, and the lengths of the tubes should be carefully adjusted, so that the end plates will be flat and without strain when riveted up.

Turning of Edges.

Edges of Ends and Angles.—The outer edge of the end plate and of the shell angle flange, also the edges of the openings for the tubes, should be neatly turned, and the mudhole opening should be neatly turned, or if drilled the edges should be chipped true.

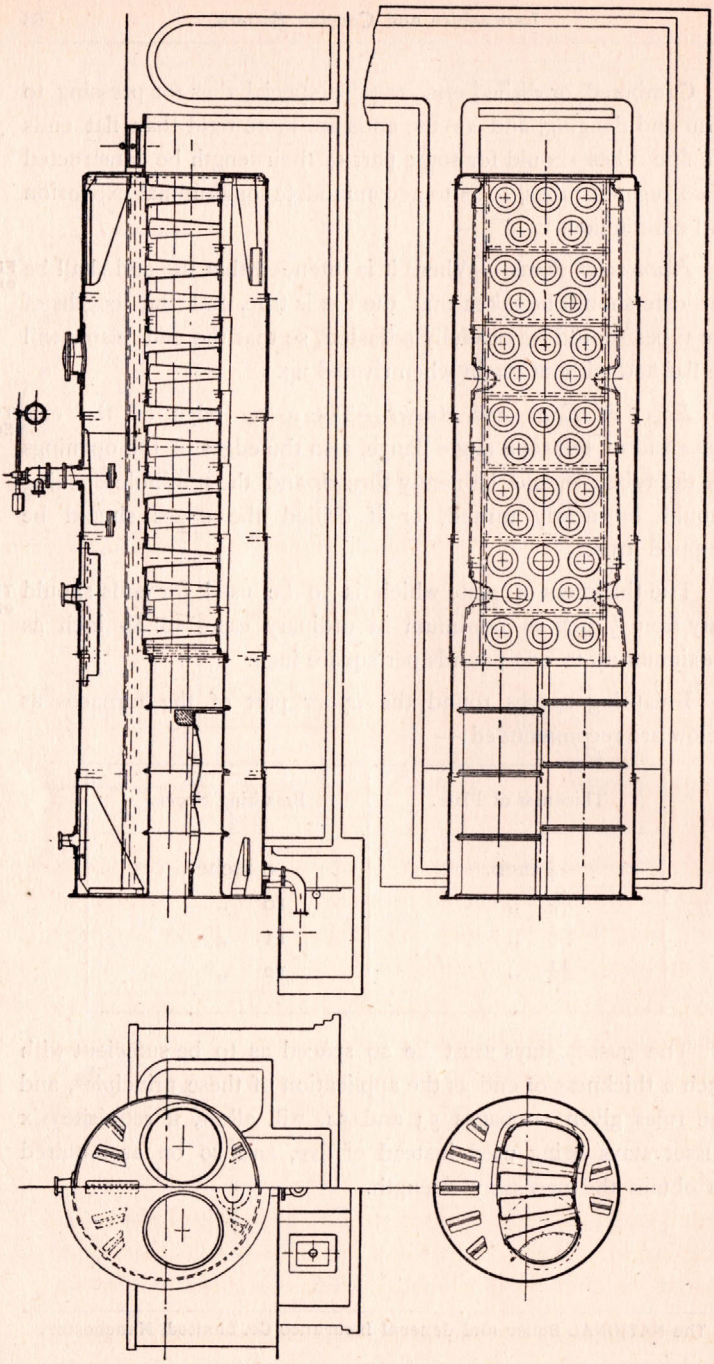
Thickness of Ends.

The thickness of plate which is to be used for ends should vary from $\frac{1}{2}$ inch as minimum in ordinary cases to $\frac{11}{16}$ inch as maximum up to 200 pounds per square inch.

Breathing spaces round the upper part of the furnaces as below are recommended:—

Thickness of Plate.	Breathing Space.
$\frac{1}{2}$ inch.	9 inches.
$\frac{9}{16}$,,	10 ,,
$\frac{5}{8}$,,	11 ,,
$\frac{11}{16}$,,	12 ,,

The gusset stays must be so spaced as to be sufficient with such a thickness of end as the application of these principles, and the rules given on pages 53 and 54, will allow, if requisite six gusset stays being used instead of five, and so on as required to obtain the necessary strength.



GALLOWAY BOILER.

VIII.

VERTICAL BOILERS.

VERTICAL BOILERS WITH CROSS TUBES.

VERTICAL MULTITUBULAR BOILERS.

EXTERNAL SHELL.

The design of the external shell to follow usual lines. As a rule, for Boilers about 3 feet diameter and over, the manhole should be strengthened by a pressed steel mouthpiece with pressed steel cover, the shorter diameter being arranged vertically.

For Boilers of a smaller diameter than 3 feet the faced mouthpiece usually would interfere with the space available inside the Boiler. In such cases, therefore, it is desirable to strengthen the manhole opening by a substantial riveted ring, the cover having a curved face instead of a flat face.

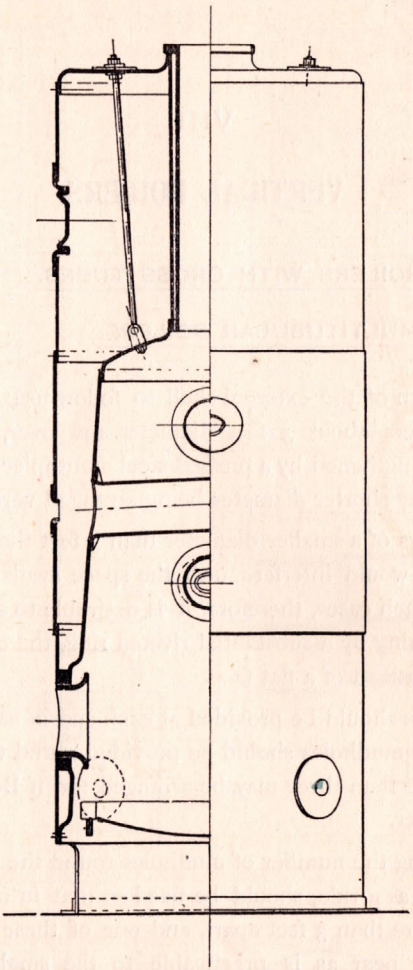
Mudholes and Sightholes.

Sight holes should be provided at each end of each cross tube. At least three mudholes should be provided round the base of the Boiler. More than three may be arranged for, if Boiler, say, over 3 feet diameter.

In deciding the number of mudholes round the bottom of the Boiler, these, as a rule, should be fixed so that in no case are the mudholes more than 3 feet apart, and one of these should always be placed as near as is practicable to the small water space between the firehole and the foundation ring. (*See* sketch, page 67.)

In all cases, as a rule, pressed steel mudhole mouthpieces and pressed steel doors should be fixed, the faces of both the mouthpiece and the door being suitably machined and the limit of the

clearance between the spigot and the mouthpiece not exceeding $\frac{1}{16}$ inch each side.

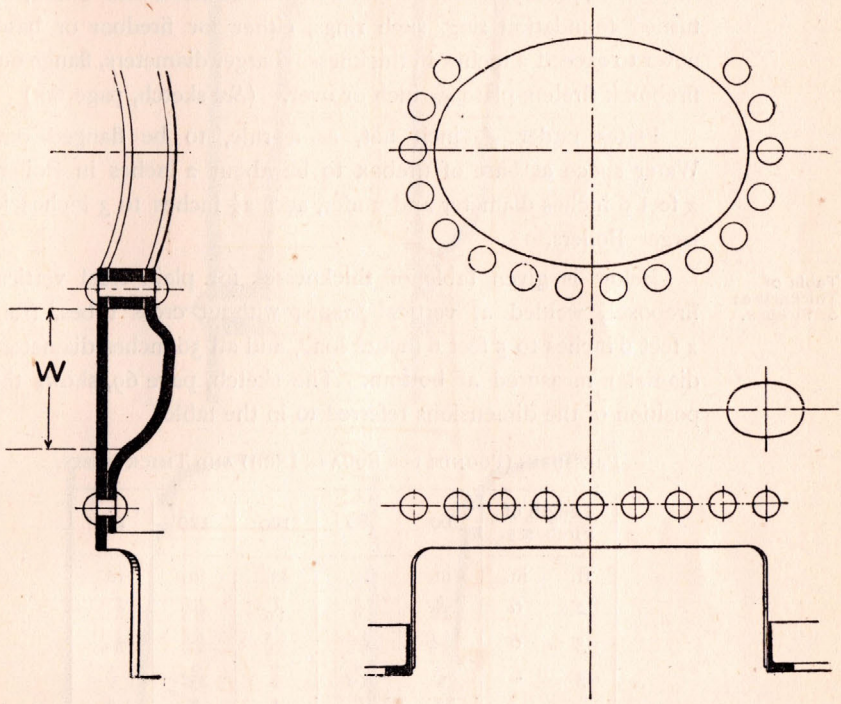


VERTICAL BOILER WITH CROSS TUBES.

Extension of Shell below Foundation Seam.

Generally the Boiler is to be so designed that the firebox shall be accessible without any digging of a pit under the firebars, and for this purpose the shell should be extended below the foundation seam, say, for a depth of 11 inches or 12 inches, and

a rectangular opening left in this extension of the shell below the firehole, say, about 12 inches deep by 18 inches wide. (*See* sketch below.) A loose door should be provided in order to cover this opening, and this loose door will serve as a damper to regulate the draught in addition to the chimney damper.



Firehole Water Space

In fixing the position of the firehole it is, as a rule, desirable that the water space (W) below should not be less than 9 inches high, to avoid choking by deposit. (*See* sketch above.)

Baseplate and access.

If the Boiler be seated on a cast iron baseplate, this casting should be provided with an opening, so that the fire side of the firebox will be accessible for cleaning and examination.

Firebar Bearers.

The firebar bearers should be arranged so that they will not retain ashes between the bearer and the firebox plates. They should be of an open pattern, so that any ashes will fall freely away, otherwise corrosion will be set up.

FIREBOX.

Firebox.

As a rule, the firebox is to be tapered, and a taper to be allowed of about $1\frac{1}{2}$ inch in diameter per 1 foot of height of firebox. Vertical seam to be welded. Firebox in one plate. Foundation seam to shell, say, Boilers 2 feet 6 inches diameter and under, by turned foundation ring, such rings, either for firedoor or base, never to exceed 2 inches in thickness. Larger diameters, flange out firebox if firebox plate $\frac{7}{16}$ inch or over. (*See* sketch, page 69.)

Plates under $\frac{7}{16}$ inch not, as a rule, to be flanged out. Water space at base of firebox to be about 2 inches in Boilers 2 feet 6 inches diameter and under, and $2\frac{1}{4}$ inches to 3 inches in larger Boilers.

Table of Thicknesses of Firebox.

Below is given table of thicknesses for plain steel vertical fireboxes, welded at vertical seams, without cross tubes, from 2 feet 6 inches to 5 feet 6 inches long, and all 36 inches diameter; diameter measured at bottom. The sketch, page 69, shows the position of the dimensions referred to in the table.

PRESSURES (POUNDS PER SQUARE INCH) AND THICKNESSES.

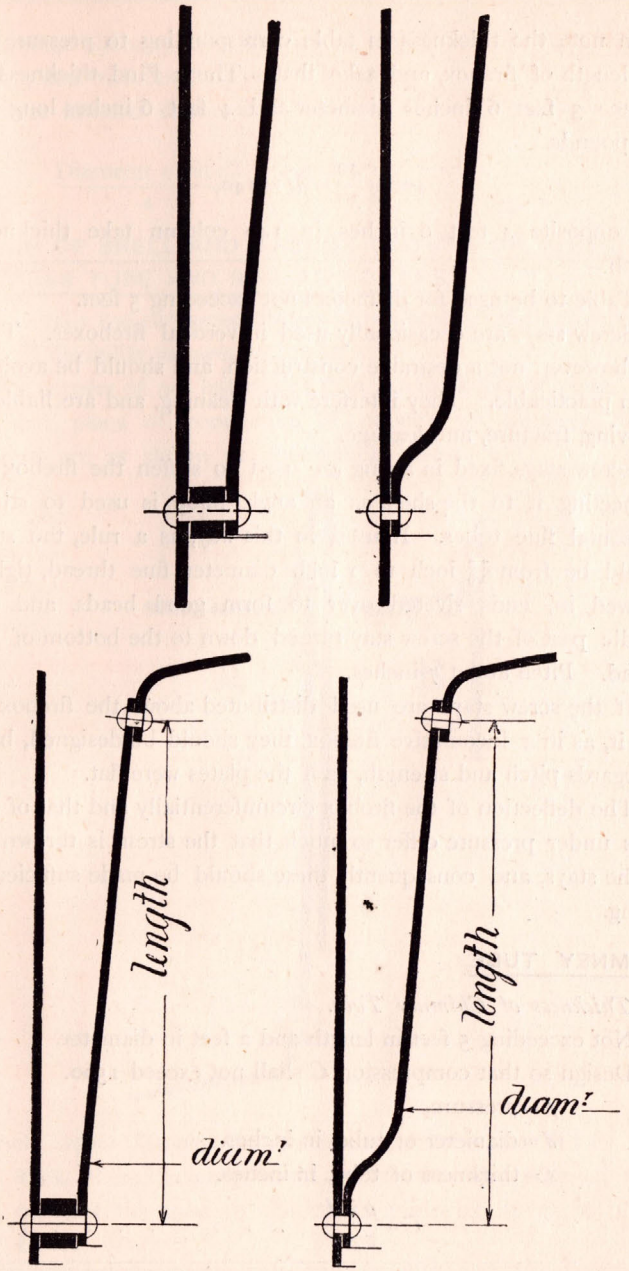
Length of Fireboxes.		60	80	100	120	140
ft.	in.	in.	in.	in.	in.	in.
2	6	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{17}{32}$	$\frac{19}{32}$
3	0	$\frac{11}{32}$	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{17}{32}$	$\frac{19}{32}$
3	6	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$
4	0	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$
4	6	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{17}{32}$	$\frac{9}{16}$	$\frac{5}{8}$
5	0	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{17}{32}$	$\frac{19}{32}$	$\frac{5}{8}$
5	6	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{17}{32}$	$\frac{19}{32}$	$\frac{5}{8}$

For other diameters than 36 inches, to find the thickness for a required pressure,

Let d = diameter in inches,

P = pressure required,

then $P \times \frac{d}{36} = M$ . . . . . (1)


length
diam.
length
diam.

Then note the thickness in table corresponding to pressure M and length of firebox, and take that. Thus: Find thickness of furnace 3 feet 6 inches diameter and 4 feet 6 inches long for 120 pounds.

$$120 \times \frac{42}{36} = M = 140,$$

and opposite 4 feet 6 inches in 140 column take thickness, $\frac{5}{8}$ inch.

Table to be used for diameters not exceeding 5 feet.

Screw Stays in Vertical Fireboxes

Screw stays are occasionally used in vertical fireboxes. They are, however, not a desirable construction, and should be avoided when practicable. They interfere with cleaning, and are liable to grooving, fracture, and leakage.

Ring of Screw Stays.

Screw stays fixed in a ring are used to stiffen the firebox by connecting it to the shell, as an angle hoop is used to stiffen horizontal flue tubes. If used in this way, as a rule, the stays should be from $\frac{13}{16}$ inch to 1 inch diameter, fine thread, tightly screwed in, ends riveted over to form good heads, and the middle part of the screw stay turned down to the bottom of the thread. Pitch about 7 inches.

Distributed Screw Stays.

If the screw stays are used distributed about the firebox to stay it, as in a locomotive firebox, they should be designed, both as regards pitch and strength, as if the plates were flat.

The deflection of the firebox circumferentially and that of the stays under pressure differ so much that the stress is thrown on to the stays, and consequently these should be made sufficiently strong.

CHIMNEY TUBE.

CHIMNEY TUBE.

Thickness of Chimney Tube.

Not exceeding 5 feet in length and 2 feet in diameter.

Design so that compression C shall not exceed 1500.

p = pressure.

d = diameter of tube, in inches.

s = thickness of tube, in inches.

$$C = \frac{p \times d}{2 \times s} \quad \ldots\ldots\ldots\ldots \quad (2)$$

Arrange chimney tube, say, 1 inch larger diameter at top than bottom, and with cast iron liner internally.

Make internal diameter of inside of liner of chimney tube about =

$$\frac{\text{Diameter of Shell}}{4} \text{ to } \frac{\text{Diameter of Shell}}{6}$$

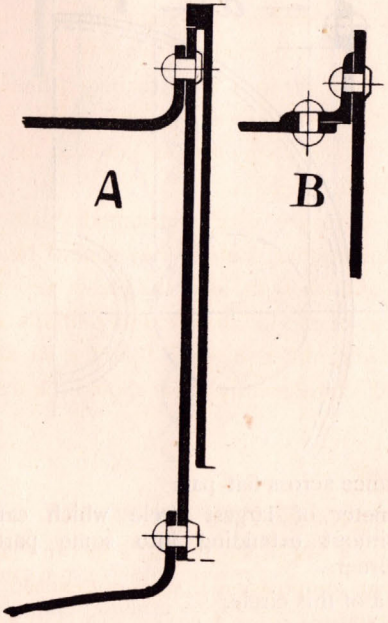
DESIGN OF SHELL AND FIREBOX CROWNS OF VERTICAL CROSS TUBE AND SIMILAR BOILERS.

SHELL AND FIREBOX CROWN DESIGN.

Generally use cambered firebox crown and flat shell crown, so as to get flexibility to meet expansion.

Use corners of not less radius than 4 × thickness of plate.

Usual plans of connection of crowns to chimney tube preferred are as shown on the sketch below.



Shell crown flanged for connection to tube as at **A**; or for small sizes, where flanging would unduly thin the edge of the plate, connect the tube to the crown plate by an angle ring, as at **B**.

Design of Crown Plates.

For diameters of Boilers up to 3 feet and up to 100 pounds per square inch, and for diameters up to 4 feet, and pressures up to 80 pounds, with internal diameter of chimney tube not less than

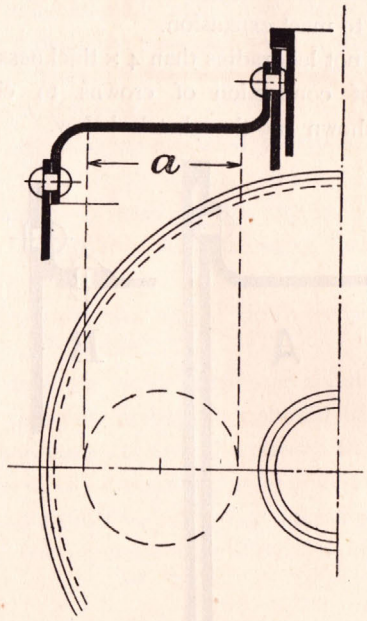
$$\frac{\text{Diameter of Shell}}{6}$$

use flat shell crown.

„ cambered firebox crown.

„ no bolt stays.

Calculate thickness of flat shell crown thus:—



a = distance across flat part

Unstayed Crown Plates.

= diameter of largest circle which can be inscribed without extending into some part of radius of corners.

Let A = area of this circle.

t = thickness of crown, in sixteenths.

p = pressure, in pounds per square inch.

Determine t by the following equation :—

$$p = \frac{200 \times t^2}{A} \quad \ldots\ldots\ldots\ldots \quad (3)$$

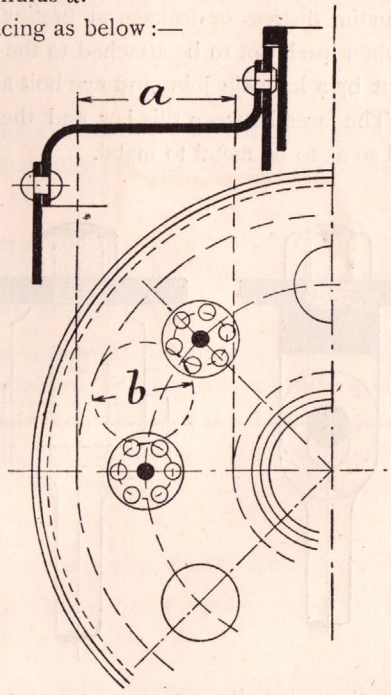
Beyond the limits mentioned above use bolt stays, generally spacing them so that circle **A** encloses them. Bolt Staying

For smaller Boilers use ordinary washers; large Boilers, riveted washers.

Generally make thickness of washers equal thickness of plate.

Make bolts sufficiently strong to carry one-half the total pressure on the flat portion of crown, namely, one-half the total pressure on annulus *a*.

Design spacing as below:—



b is diameter of largest inscribed circle on flat part passing through centres of rivets of washers, or through centres of bolts if no riveted washers.

B = area, in square inches, of circle *b*.

t = thickness of crown, in sixteenths.

p = pressure, in pounds per square inch.

Determine t by the following equation:—

$$p = \frac{200 \times t^2}{B} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

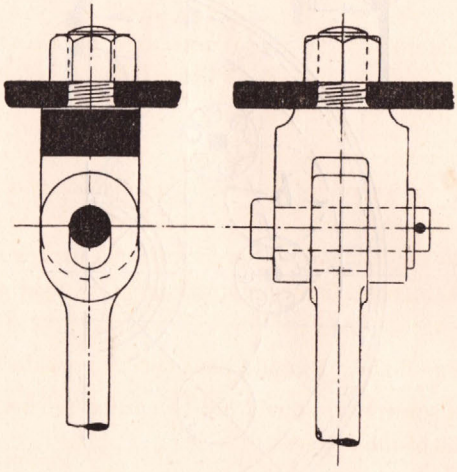
CAMBERED CROWNS.

Cambered Crowns.

The thickness of a cambered crown should be not less than that given in "Cambered Ends" (*see* pages 33 and 34), and generally should exceed the thickness of the shell or furnace to which it is attached by $\frac{1}{16}$ inch to $\frac{1}{8}$ inch. Radius of camber, as a rule, to be equal to diameter of the cylindrical shell to which the plate is connected.

Attachment of Bolt Stays to Cambered Crowns.

To avoid undue distress or leakage at firebox end, as a rule, vertical stays, when used, not to be attached to the firebox end by double nuts, but by a knuckle joint and eye bolt attached to the jaw by a pin. The faces between this jaw and the firebox crown to be machined so as to be metal to metal.



The slots in the eye bolts not to be circular, but elongated vertically, so that if the firebox crown should rise owing to expansion it would not push the vertical stays upwards, and so distress the shell crown.

IX.

LOCOMOTIVE TYPE BOILERS.

FIREBOX.

FIREBOXES.

In arranging the staying and the proportions of the firebox special regard should be paid to considerations of freedom of expansion.

If the water space be too narrow, say, less than $2\frac{1}{2}$ inches at any point, it is easily choked by incrustation; and also the stays, being very short, are not flexible.

It is desirable that the water space about a locomotive firebox should not anywhere be less than $2\frac{1}{2}$ inches, and generally an endeavour should be made to make the water space not less than $2\frac{1}{2}$ inches at the foundation, increasing to 4 inches at the top of the box, this increase being obtained either by tapering the firebox or the casing.

Very large fireboxes should have special consideration.

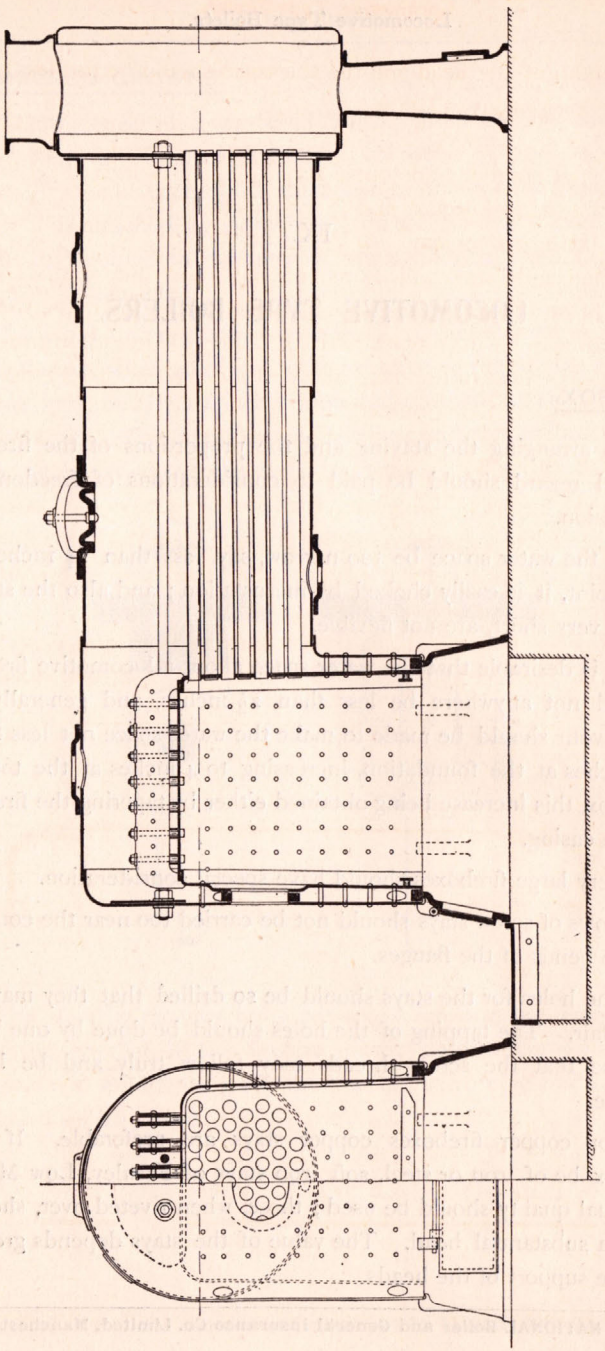
Rows of screw stays should not be carried too near the corners or the bends of the flanges.

Screw Stays of Locomotive Boilers, Combustion Chambers, and for similar purposes.

The holes for the stays should be so drilled that they may be truly fair. The tapping of the holes should be done by one long tap, so that the screw threads may follow truly and be both in line.

Locomotive Fireboxes of Copper.

For copper fireboxes copper stays are preferable. If the firebox be of iron or steel, soft iron stays of Farnley, Low Moor, or equal quality should be used; these, when riveted over, should have a substantial head. The value of the stays depends greatly on the support of the heads.



LOCOMOTIVE TYPE BOILER.

The lip of the head and the thickness should be not less than the dimensions given below :—

Material.	Lip of Head.	Thickness of Head.
Copper firebox and copper stay	$\frac{3}{16}$ inch	$\frac{1}{4}$ inch
Iron or steel firebox and $\frac{3}{4}$ inch stay...	$\frac{1}{16}$,,	$\frac{3}{32}$,,
iron or steel stay ... $\frac{7}{8}$,, ,, ...	$\frac{3}{32}$,,	$\frac{1}{8}$,,

Nuts on Screwed Stays.

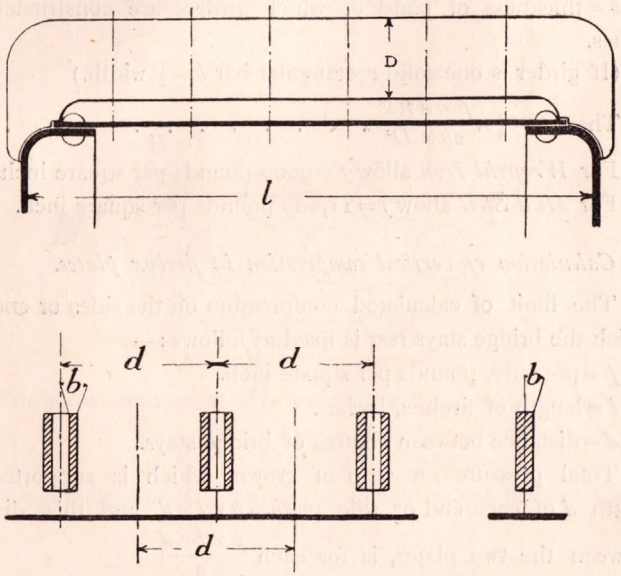
Where nuts are applied on screwed stays they should be a good fit on the screw, and should bed truly on the plate, so that the nut may be in contact with the plate all round, in order to conduct the heat to the plate and prevent the nut itself and the end of the stay being burned.

BRIDGE STAYS.

BRIDGE STAYS.

In the design of bridge stays without sling stays it is assumed that the bridge stays transmit the pressure from the strip of crown supported to the vertical plates below the ends of the bridge stays, causing compression in these vertical plates.

In order that the bridge stays may transmit the downward force to the vertical plates it is necessary generally that the



curvature between the side and the crown, or the end and the crown, should not be too large, and also that the toes of the bridge stays should be very carefully fitted to bed over the corner and to transmit the force to the vertical plate. The toes should be part of the bridge, and not a separate piece attached to it.

Bridge stays should be made out of one solid piece of wrought iron or mild steel, suitably enlarged at the bolt holes and fitted at the toes, or the bridge stay should be of two plates properly connected together and shaped at the toes to bed truly.

Ordinary bridge stays for crowns of locomotive fireboxes.

For designing bridge stays without slings the following formula is recommended :—

p = pressure, pounds per square inch.

l = length of firebox spanned by girders, in inches. (*See* diagram, page 77.)

d = distance between centres, *i.e.*, pitch.

D = depth of girders, in inches.

b = thickness of plate of which girders are constructed, in inches.

(If girder is one solid rectangular bar $b = \frac{1}{2}$ width.)

Then $f = \frac{3}{4} \times \frac{p \times d \times l^2}{2b \times D^2}$ . . . . . (1)

For *Wrought Iron* allow $f = 9000$ pounds per square inch.

For *Mild Steel* allow $f = 11{,}000$ pounds per square inch.

Calculation of vertical compression in firebox plates.

Compression on Supporting Plates.

The limit of calculated compression on the sides or ends on which the bridge stays rest is fixed as follows :—

p = pressure, pounds per square inch.

l = length of firebox, inches.

d = distance between centres of bridge stays.

Total pressure on strip of crown, which is supported by length d of each end or side plate $= p \times l \times d$, and this, divided between the two plates, is for each $\frac{p \times l \times d}{2}$

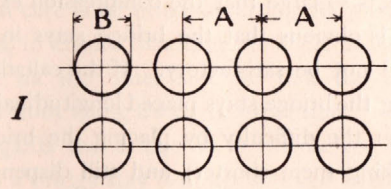
Let s = thickness of firedoor plate, inches.

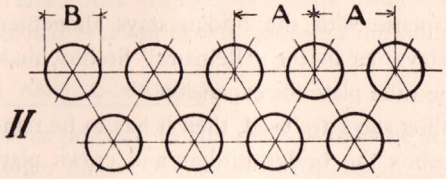
S = thickness of tube plate, ,,

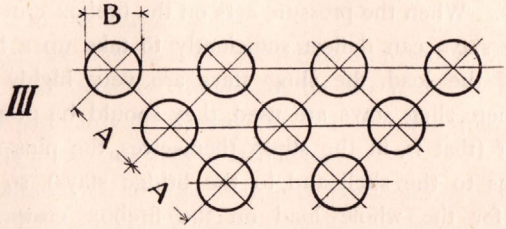
B = outside diameter of tube, ,,

A = pitch of tubes, ,,

With tubes arranged as I. or II. measure pitch A horizontal.







With holes arranged as III. measure pitch A along diagonal.

$$K = \frac{A - B}{A}$$

then the compression C (pounds per square in.) on firedoor plate

$$= \frac{p \times l \times d}{2 \times d \times s} = \frac{p \times l}{2 \times s} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and compression (pounds per square in.) on the tube plate

$$= \frac{p \times l \times d}{2 \times K \times S \times d} = \frac{p \times l}{2 \times K \times S} \quad \ldots\ldots\ldots \quad (3)$$

For copper the compression allowable $C = 4000$ pounds per square inch.

For good furnace iron or furnace mild steel C = 14,000 pounds per square inch.

When ordinary Bridge Stays inapplicable.

When the firebox is so large that the compression exceeds the limits given, then it is obvious that the bridge stays in question by themselves would not be satisfactory. If the calculation has been made supposing the bridge stays placed longitudinally, it may be possible to meet the difficulty by placing the bridge stays transversely (so making them shorter), and still dispensing with sling stays. Where, however, this expedient still results in the compression exceeding the limits fixed, then, as a rule, it is preferable to dispense with the bridge stays altogether and to adopt screwed stays, as in the "Belpaire" firebox, making due provision near the tube plate for expansion.

"Belpaire" Staying.

SLING STAYS.

If, however, sling stays are used, then it has to be remembered that in a long firebox the bridge girders and crown plate form a comparatively flexible arrangement, and under equal loads the deflection of the ordinary bridge girders and the firebox crown is very much more than the extension which could take place in the sling stays. When the pressure acts on the firebox crown, before the bridge stays can deflect sufficiently to take up a large proportion of the load, the sling stays are very highly stressed. Hence, where sling stays are used, they should be proportioned throughout (that is, in the slings themselves, the pins and their connections to the shell and to the bridge stays), so as to be adequate for the whole load on the firebox crown. Where such stays are adopted and proportioned in this way, then in designing the bridge girders use equation (1) (page 78) and proportion the bridge girder as if there were no slings, except that f, as below, should be used instead of the values given with equation (1):—

Iron $f = 25{,}000$
Steel $f = 30{,}000$

In the foregoing no account has been taken of the differences of relation which arise owing to the differences of expansion upwards of the firebox and firebox casing. No doubt this affects the question in some degree, but practically large firebox crowns are so flexible that the work of supporting them is bound to fall on to the sling stays, and these should, therefore, be proportioned accordingly.

Special care is required in fitting the slings and pins together when cold, to ensure that they are tight in tension with a suitable allowance on the other side, so that the firebox shall not be prevented from rising owing to expansion.

As regards the bridge stays themselves, it is to be observed that when a bridge stay is firmly screwed to the firebox crown, with ferrules between the undersides of the bridge girder and the crown, the crown and girder will to some extent act as a trussed structure, but no account is taken of this feature in the calculations.

ACCESS AND CLEANING.

Access to Interior.

Whenever practicable, the Boiler should be arranged so that a man can enter the inside above the tubes and pass to the firebox end inside. To enable this to be done, the number of longitudinal stays should be as few as possible, the ends being suitably stiffened so that the pressure on the ends may be properly transmitted to the stays.

Manholes.

In larger sizes the manhole should be arranged, if possible, in the barrel. If, however, the Boiler be too small for a man to enter, then preferably the manhole should be in the firebox casing, immediately above one side of the firebox.

Mudholes.

There should be at least four oval handholes or mudholes at the bottom of the firebox casing to enable the deposit there to be cleaned out, and also to allow of the stays and plates adjacent to these openings being properly examined. These handholes should be in the rounded corner, with the major axis vertical, thus enabling the plates and stays at each side of the corner to be examined. This arrangement, of course, necessitates the corners being made to a large radius.

Two handholes should also be provided near the upper part of the firebox, so that the upper stays can be examined. Provision should also be made for the proper cleaning and examination of the furnace crown.

Access to Firebox.

The foundation casting or seating underneath the firebox should be provided with an opening, so that the fire side of the firebox may be accessible for cleaning and examination.

Firebar Bearers.

The firebar bearers should be arranged so that they will not retain ashes between the bearer and the firebox plates. They should be of an open pattern, so that any ashes will fall freely away, otherwise corrosion of the plates will be set up.

X.

MARINE DRY-BACK BOILERS AND SIMILAR TYPES.

Access Internally.

In the design of these Boilers special consideration should be given to access internally, so that the inspector and boiler cleaners may pass freely between the nests of tubes, and also have a reasonable amount of space allowed between the furnace tubes and the undersides of the smoke tubes. Generally, the arrangement of smoke tubes in parallel nests facilitates this.

Undue crowding of tubes in order to secure heating surface should be avoided.

Cross tubes in such Boilers are objectionable, as they cannot be cleaned.

Flue Tubes.

Generally, it is desirable to reduce the flue tube by about 6 inches in diameter in the ring which is connected to the back end plate, or in the preceding ring.

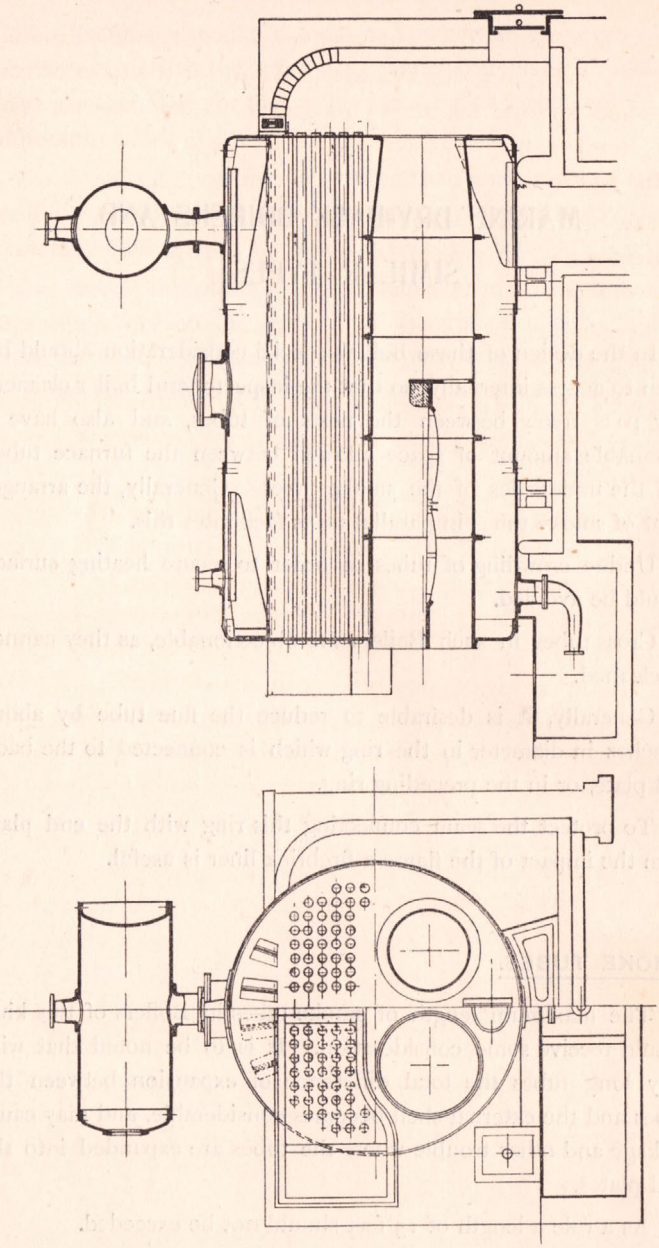
To protect the seam connecting this ring with the end plate from the impact of the flame a firebrick liner is useful.

SMOKE TUBES.

Length of Smoke Tubes.

The maximum length of smoke tubes in Boilers of this kind should receive some consideration. It is to be noted that with very long tubes the total difference of expansion between the tubes and the external shell becomes considerable, and may cause leakage and other trouble where the tubes are expanded into the end plates.

As a rule a length of 15 feet should not be exceeded.



MARINE DRY-BACK BOILER.

Expanding and Beading of Smoke Tubes.

One objection to the beading of tubes is that, unless carefully done, it is liable to spring the expanded parts of the tubes off the plates and to cause the tubes to be leaky. If well and carefully done, however, it is an advantage to bead the tubes in addition to expanding them.

The best job of expanding the tubes is made when the scale is carefully filed or ground off the tubes and the holes carefully cleaned, so that a metal to metal joint is made.

Ferruling.

Ferruling at the hottest end is advantageous in addition to beading.

Tubes of a moderate thickness are better than thin tubes, because they maintain their expanded joints better and have a greater margin for wear and tear.

Stay Tubes

Stay tubes are sometimes nutted at the ends, or simply expanded or beaded over where they pass through the plate, but in all cases such stay tube ends require screwing at each end, and they should be carefully screwed into both plates at the same time, and then expanded up so that the screws bed firmly at each end. In order that this may be done accurately, usually the plates require tapping with one long tap. Stay tubes in all cases require fixing before the others are put in, so that the nuts can be firmly screwed home. The nuts of the stay tubes should not, as a rule, be thicker than the plate through which the tubes pass.

XI.

TUBES OF WATER-TUBE BOILERS.

WATER TUBE.

Solid drawn mild steel tubes are recommended. It is desirable that at least the rows of tubes nearest the fire should be solid drawn.

Each tube should be carefully and evenly expanded into the hole, and the projecting part should be bell-mouthed, to safeguard against drawing out.

Bell-mouthing.

The tubes should be sufficiently ductile to stand such expanding and bell-mouthing without splitting or showing other flaws. If tubes are welded the welds should be turned away from direction of flow of the hottest gases.

Bell-mouthing of Water Tubes and large Smoke Tubes.

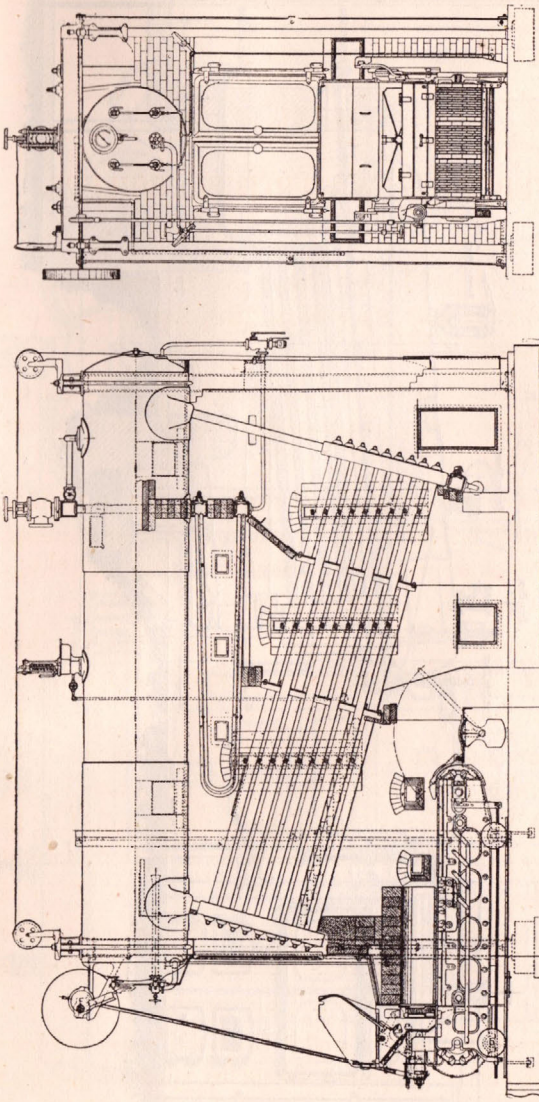
When tubes are bell-mouthed, in order to increase their hold in the plate in which they are expanded, generally the bell-mouthing should be as follows :—

The tube end should project beyond the plate into which it is expanded from $\frac{1}{4}$ inch to $\frac{1}{2}$ inch. If the projection is less than $\frac{1}{4}$ inch the tubes cannot be properly bell-mouthed. If it is greater than $\frac{1}{2}$ inch the ends are liable to be split in forming the bell-mouth.

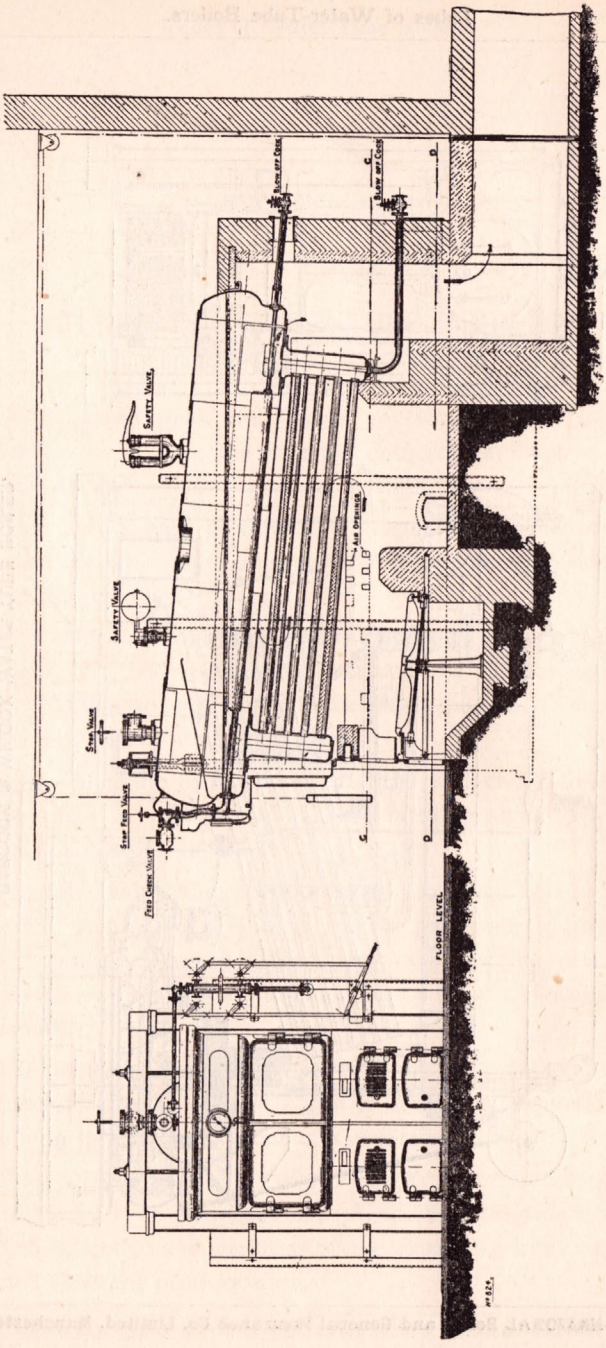
Bell-mouthing should be such that the bell-mouthed end of the tube shall be about $\frac{1}{8}$ inch greater in external diameter than the hole in the plate into which it is expanded.

HYDRAULIC TESTS.

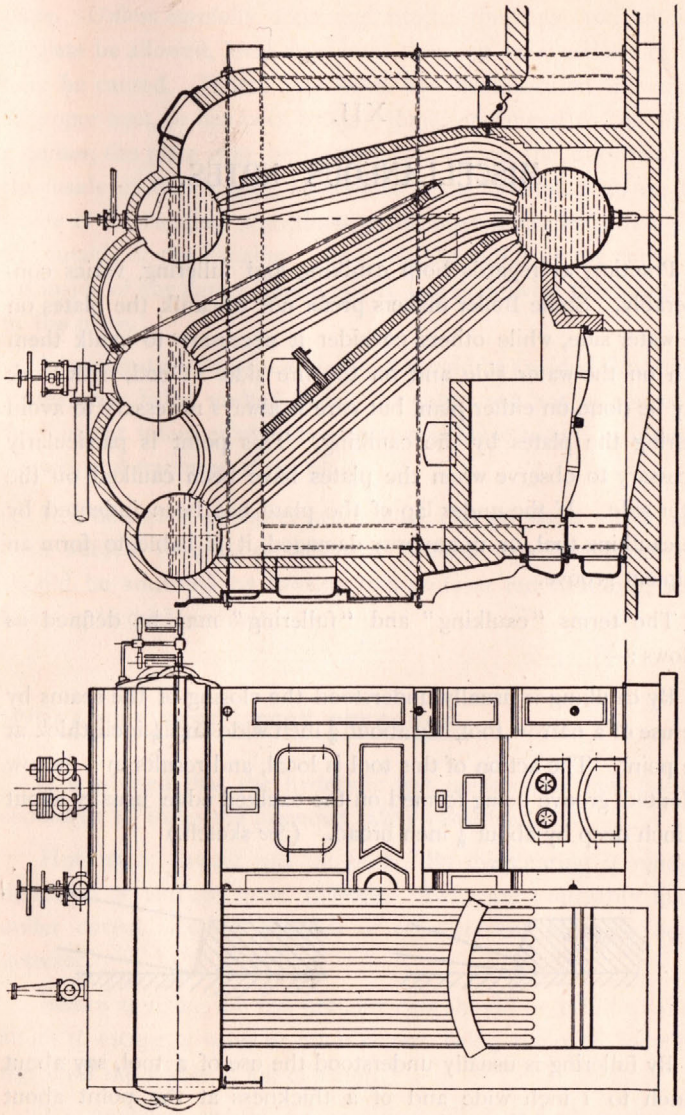
When such Boilers are under hydraulic tests on completion they should not be supported except by slings or girders above the Boilers, so that any weakness of the expanded joints may be detected when the pressure is on.



BABCOCK & WILCOX WATER-TUBE BOILER.



MARSHALL WATER-TUBE BOILER.



STIRLING WATER-TUBE BOILER.

XII.

MISCELLANEOUS NOTES.

Caulking and Fullering.

Practice, as regards both caulking and fullering, varies considerably. Some Boiler makers prefer not to caulk the plates on the water side, while others consider it necessary to caulk them both on the water side and on the fire side. Good, tight work may be done on either plan, but care is always necessary to avoid injuring the plates by the caulking. This point is particularly necessary to observe when the plates have been caulked on the water side. If the under lap of the plate has been indented by the caulking tool, or in any way damaged, it is liable to form an incipient groove.

The terms "caulking" and "fullering" may be defined as follows :—

Caulking.

By caulking is usually understood the closing of the seams by the use of a narrow tool, say about $\frac{3}{4}$ inch wide and $\frac{1}{8}$ inch thick at the point. The action of this tool is local, and results in a narrow indented groove being formed on the caulked edge, usually about $\frac{1}{16}$ inch deep by about $\frac{1}{8}$ inch broad. (*See* sketch.)

Fullering.

By fullering is usually understood the use of a tool, say about $\frac{3}{4}$ inch to 1 inch wide and of a thickness at the point about equal to the thickness of the plate. (*See* sketch.) The use of this tool results in the fullered edge being smooth right across.

Flanging generally.

The operation of flanging usually involves stretching of the plate. Unless carefully done, and proper thickness and amount of plate be allowed, local or general thinning to a serious extent may be caused. Where this work has been carried out at an improper heat, or has been roughly done, or flanged to too sharp a corner, the plate may be fractured either on the outside or on the inside of the flanging, sometimes being crumpled up on the inside of the flanging, sometimes small, open fractures showing on the outside of the flanging. As a rule the radius of the corner of the inside of any flange should not be less than twice the thickness of the plate.

Blue Heat.

Working steel plates at a blue heat should be avoided altogether. Plates when being worked should be either red hot or else cold. Working at intermediate temperatures is liable to set up internal stresses and fractures of the plate.

Annealing.

It is desirable, whenever practicable, that all plates which have been welded, flanged, or otherwise worked in a hot condition, should be annealed by heating to a uniform red heat, and then allowing to cool gradually.

HYDRAULIC TEST.—TEST PRESSURE.

Hydraulic Test.

It is advised that the hydraulic test pressure for new Boilers should be $1\frac{3}{4}$ times the intended safety valve load.

Hot water should not be used. By evaporating it renders detection of leakage more difficult. Tests are preferably made under cover; outside, showers of rain prevent leakages being detected.

Application of Test Pressure.

Before applying the test pressure care should be taken to allow all air to escape, a valve or other covers being removed to effect this until the Boiler is full of water.

The pressure should be steadily applied, and when the test pressure is reached it should be steadily maintained for not less than half an hour.

The necessary observations and examination before, during, and after the test should be made by the "National" Inspector.

Test on Seat. It is desirable that after a new Boiler has been placed on its seat it should again be subjected to a hydraulic test, in order to ascertain whether it has been strained in transit. For this test a pressure equal to the safety valve load is usually sufficient in the case of Lancashire, Cornish, and similar Boilers.

XIII.

BOILER FITTINGS.

SAFETY VALVES.

SAFETY VALVES.

For large Boilers, such as ordinary sizes of Lancashire Boilers and Cornish or Water-tube Boilers, it is recommended that each Boiler should have two safety valves.

Large Boilers.

The combined area of the two safety valves should not be less than the area obtained from the table which follows.

For pressures over 100 pounds, a convenient arrangement is to make one of the valves of the spring type enclosed, provided with a test lever and suitable waste pipe. The other valve to be an open one of the external dead-weight pendulous type; this valve to be loaded to about 5 pounds above the pressure of the enclosed valve.

With this arrangement the blowing off in the ordinary way is done by the enclosed valve, avoiding the annoyance of steam escaping into the boiler-house, while the pendulous safety valve remains as a check on the efficiency of the other, ready for action in case of necessity.

TABLE OF SAFETY VALVE AREAS.

P = Boiler pressure, in pounds per square inch.

A = Area of safety valve, in square inches, per square foot of firegrate.

(These are Board of Trade figures.)

P	A	P	A
60	·5	135	·25
65	·468	140	·241
70	·441	145	·234
75	·416	150	·227
80	·394	155	·220
85	·375	160	·214
90	·357	165	·208
95	·34	170	·202
100	·326	175	·197
105	·312	180	·192
110	·300	185	·187
115	·288	190	·182
120	·277	195	·178
125	·267	200	·174
130	·258		

Safety Valve Connections and Waste Pipes.

In all cases it is desirable that at least one of the safety valves should be connected direct to the main steam space of the Boiler by a separate connection used for the safety valves only. The attachment of a safety valve and junction valve to the same branch is very objectionable

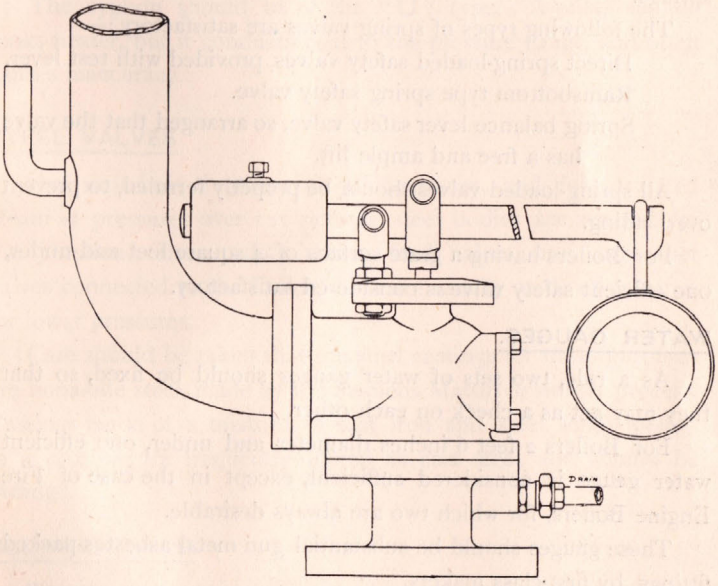
Where a Boiler is provided with a steam dome, sometimes the safety valve is connected to the steam dome. This arrangement will usually be a satisfactory one, provided that there is no anti-priming pipe intervening between this steam dome and the steam space of the Boiler.

It has to be remembered that when Boilers prime, and there is any mud in the Boiler, it is liable to be carried on to the anti-priming pipe and choke up the openings in this pipe. If, then,

the safety valves were on the other side of the anti-priming pipe, the valves might be rendered inoperative in this way.

Waste Pipes for Enclosed Safety Valves.

Waste pipes for enclosed safety valves should in all cases have a small sentinel pipe (*see* sketch) fixed on the waste pipe near to the valve, so that the blowing off of the valve may be shown by steam escaping from the sentinel pipe. In addition, if the waste pipe rises up to the roof, or above the level of the valve, a drain should be provided at the bottom of the pipe, so that water may not lodge on top of the safety valve.



If there are a number of Boilers in a range each provided with low-water safety valve, it is desirable that the waste pipes for each of these safety valves should be separate; that is to say, there should not be branch waste pipes connected to one common main waste pipe. The latter arrangement is open to the objection that in case one of the Boilers be at rest the low-water safety valve will be open, owing to the action of the weight of the float causing the valve to lift. If, under these circumstances, one of the other valves connected with the same common waste pipe were to blow,

possibly some of the steam might be forced into the Boiler which is at rest, and cause scalding of the boiler cleaners or inspector.

Types of Safety Valves.

In ordinary sizes of Vertical Boilers or Locomotive-type Boilers, as a rule, two safety valves on a "Y" pipe, connected to a separate branch on the crown, should be provided.

For fixed Boilers, safety valves of the pendulous dead-weight type or knife-edge lever-and-weight type are recommended.

For moving Boilers, the safety valves should be loaded by springs.

The following types of spring valves are satisfactory :—

Direct spring-loaded safety valves, provided with test lever.

Ramsbottom type spring safety valve.

Spring balance lever safety valve, so arranged that the valve has a free and ample lift.

All spring-loaded valves should be properly ferruled, to prevent overloading.

For Boilers having a grate surface of 4 square feet and under, one efficient safety valve is considered satisfactory.

WATER GAUGES.

WATER GAUGES.

As a rule, two sets of water gauges should be fixed, so that they may act as a check on each other.

For Boilers 2 feet 6 inches diameter and under, one efficient water gauge is considered sufficient, except in the case of Fire Engine Boilers, for which two are always desirable.

These gauges should be substantial gun-metal asbestos-packed fittings, by first-class makers.

Each gauge glass should be provided with a suitable protector.

Water Gauges on Water-tube Boilers.

As the gauge glasses in Water-tube Boilers are at a considerable height from the firing floor, bursting of the glass is not likely to seriously injure anyone; as, however, men sometimes have to work near the water gauges, protectors should be provided, and these should be of the swing-back pattern, so that when not required they may be swung back and give a clear and unobstructed view of the gauge glass.

Generally the water gauges should be provided with suitable handles and chains, or links, so that the whole of the testing can be satisfactorily done by the attendant from the firing floor.

In high Boilers of this class it is frequently best to provide a platform at a suitable height above the firing floor, giving access to feed valves and water gauges. Such platforms should be substantially constructed of open bar work, so as not to seriously obstruct the light.

Pressure Gauge and Syphon.

Only the best makes of pressure gauges are advised.

The syphon should be of the "U" type. A pillar syphon looks neater, but it conducts heat to the pressure gauge, and often causes inaccuracy.

STEEL VALVES

Steel Valve Bodies.

For superheated steam at all pressures, and for saturated steam at pressures over 120 pounds, steel bodies are advised for the junction valves, safety valves, feed valves, and any other valves connected to the Boiler which are usually made of cast iron for lower pressures.

Care should be taken that the steel castings for these purposes are bona-fide steel, made by the Siemens-Martin or similar process. Castings made of a mixture of cast iron and steel scrap melted in a crucible are frequently offered as cast steel, and should be avoided.

BLOW-OUT PIPE.

Blow-out Pipe.

The elbow pipe connecting the blow-out tap to the stand pipe in Lancashire and Cornish Boilers in all cases to be of cast steel. The object of adopting steel is to obtain a pipe which is tough and ductile, and it is not with a view of obtaining a high tensile strength. Cast iron pipes are quite satisfactory so far as regards tensile strength, but they are unsatisfactory as regards their brittleness and want of ductility.

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